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Heating · Piping and Air Conditioning

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JOURNAL SECTION OF THE A.S.H.V.E.

AUGUST, 1931

HIGH TEMPERATURE STEAM PIPING: DESIGNING OFFICE BUILDINGS; SAVING MONEY IN MOTOR OPERATION; DISCUSSION ON EXHAUST SYSTEMS » » » » »

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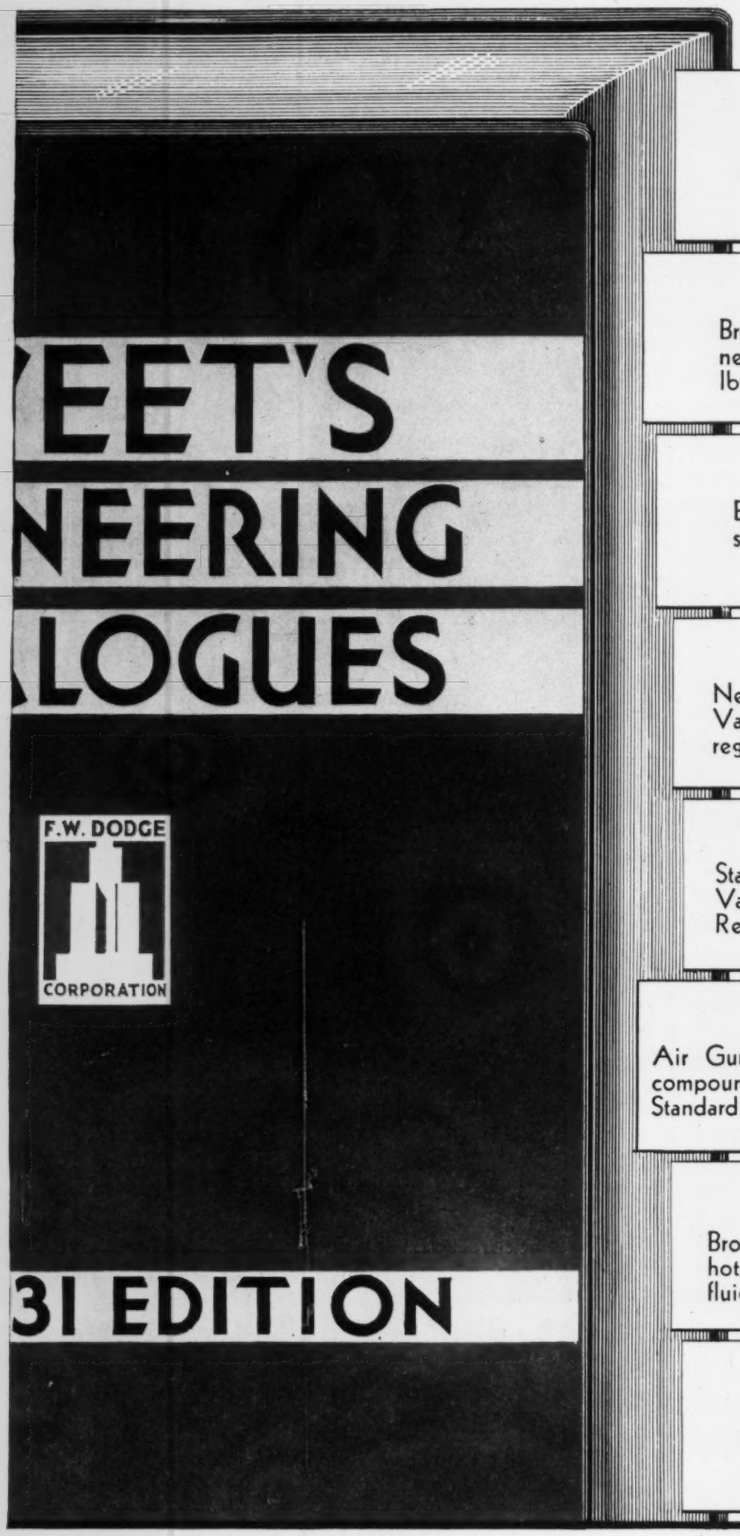

FIG. 890
Jenkins Iron Body Bronze Mounted Globe Valve, Screwed. Regrinding, union bonnet.

FIG. 750
Bronze Regrinding Union Bonnet Globe Valve. For 200 lbs. steam working pressure.

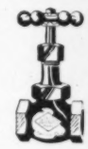

FIG. 357
Bronze Selclo Valve. For severe service where valve is operated hurriedly or carelessly.

FIG. 743
New Standard Bronze Globe Needle Valve, with bronze spindle. For close regulation of air, gasoline, and other fluids.

FIG. 124
Standard Bronze Y or Blow-Off Valve. Opening in line with pipe. Renewable disc and seat ring.

FIG. 710
Air Gun. Renewable Disc specially compounded for compressed air service. Standard, flat, rounded or extension tip.

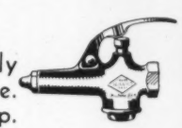

FIG. 720
Bronze Rapid Action Valve for hot and cold water and other fluids. Quick flow, on and off.

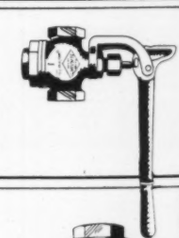

FIG. 762
Jenkins Bronze Regrinding Swing Check Valve, Screwed. Renewable bronze clapper.

FOR your convenience we are publishing in the 1931 edition of Sweet's Engineering Catalogues the complete Jenkins BOOK OF VALVES. You will find it on pages 819 to 866. Consult this catalogue on your valve requirements. Note particularly that it contains a large number of special designs, a few of which are illustrated above, designs that are finding a wide use in a diversity of industrial application

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"The cost of heating the largest office buildings is around $100,000 a year. Anything that will reduce the heat loss from such buildings by twenty-five per cent is well worth serious consideration."—*Sterling S. Sanford.*

"The station . . . offers a water piping design which not only demonstrates the latest practice, but which incorporates many features and details of interest and value to the designer of an industrial pumping plant"—*Orville B. Carlisle and Duane E. Kennedy.*

"The air conditioning and heating engineer has a certain amount of electrical equipment under his supervision, and must be familiar to some extent with the fundamentals of its operation."—*Louis Mackler.*

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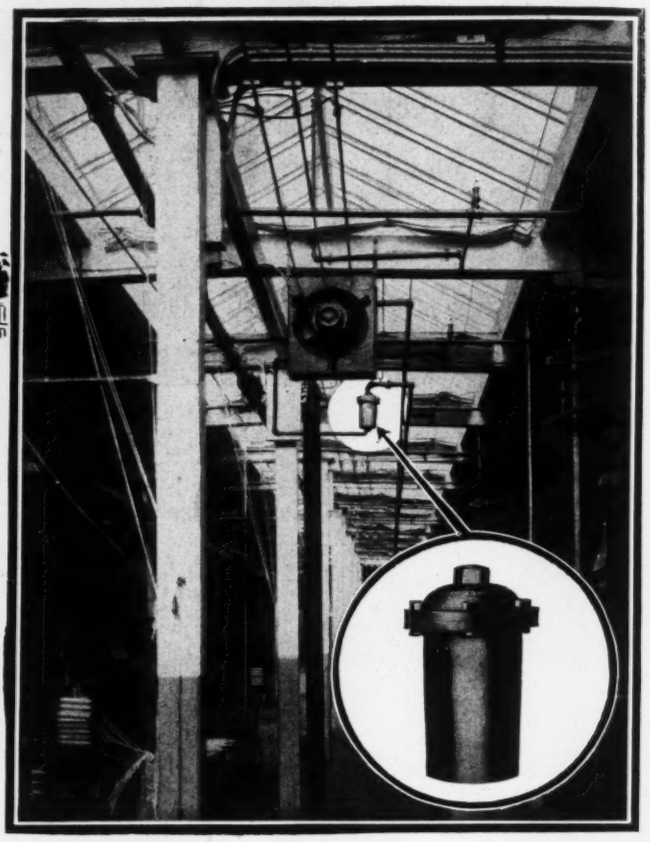

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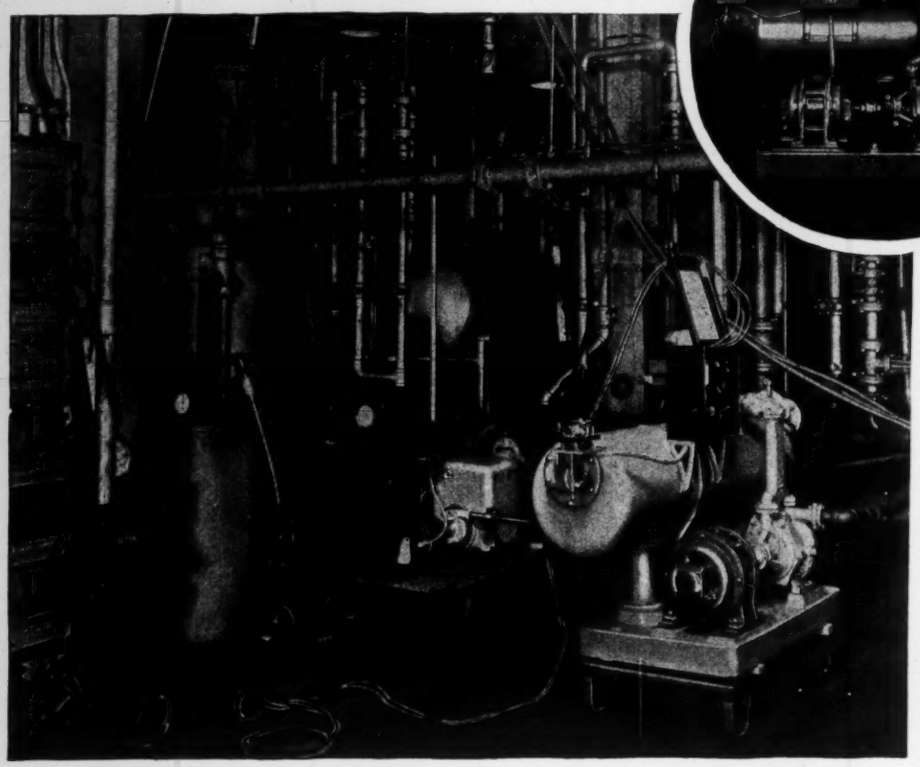

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Heating · Piping and Air Conditioning

Vol. III August, 1931 No. 8

Design Controls Cost of Heating

Buildings Should be Planned to Allow Heating Economies

By Sterling S. Sanford*

DESIGN of a building divides itself naturally into two parts, architectural and mechanical. The latter pertains to all of the mechanical equipment, including the heating and air conditioning systems. The two designs must, of course, be carefully coordinated to produce a building which can be heated with economy and with the proper degree of comfort.

Too often buildings are designed without sufficient regard for the problem of heating them. In some cases, features of architectural design make the building difficult to heat comfortably or economically; in other cases, particularly in buildings heated by fan systems, the heating system is hampered by a lack of space allotted to it. If more architects were thoroughly familiar with the heating engineer's problems, there would be less chance of the architectural design interfering with the proper and economical heating of the building.

Cutting Down Heat Loss

It is not difficult to find examples of heat being wasted because of improper design or construction. A building

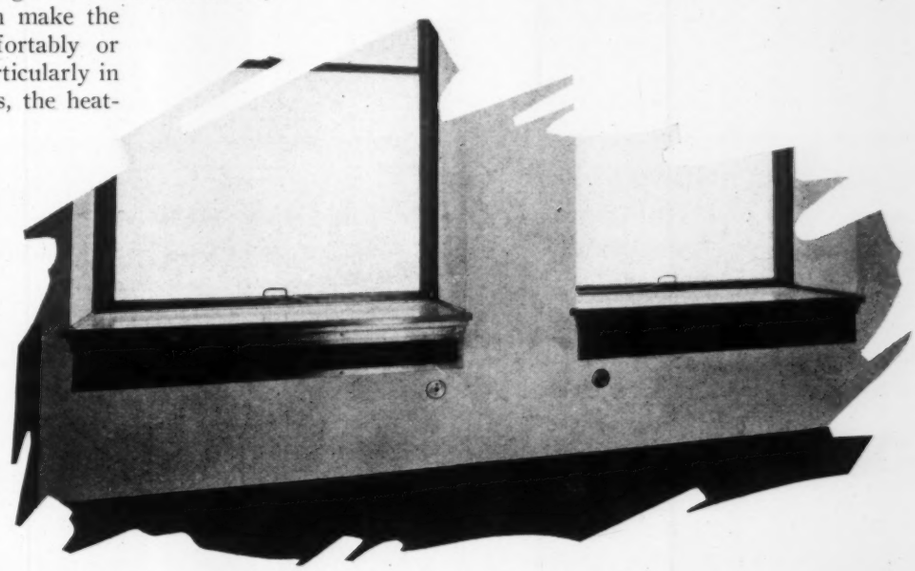


FIG. 1—RADIATOR ENCLOSURES UNDER WINDOWS. IT IS PROBABLE THAT IF THE GRILLES HAD BEEN PLACED IN THE TOPS OF THE ENCLOSURES WINDOW DRAFT COULD HAVE BEEN ELIMINATED

* Heating engineer, The Detroit Edison Co., Detroit, Mich.

The author is thoroughly familiar with the economical use of steam in office buildings. How the proper architectural and mechanical design of a modern skyscraper can do much to promote operating economies in the years to come is told in these "experience notes".

loses heat in two ways, by transmission losses through the building materials, that is through the walls, glass, and roof, or by the infiltration of cold outdoor air which causes the loss of an equal volume of warm air from the building. In a large office building, these losses are divided about as follows:

Transmission losses through walls and roof	30%
Transmission losses through window glass	40%
Infiltration losses	30%

These proportions will vary, of course, according to the type of construction. If the building is well constructed and has thick walls—as ordinarily found in office buildings—the heat transmission through the walls is usually so low that heat insulation cannot be economically justified. For walls of lesser quality and for roofs, insulation can be used to good advantage. If the walls are built of hollow tile, the tiles should be laid so that air circulation through the wall is prevented as much as possible. The infiltration losses occur mainly around the windows. Of the total heat losses from the building, the infiltration and transmission losses through windows make up about 70 per cent. It would seem that effort to reduce heat losses should be concentrated on the windows.

Windows Should Be Tight

It is most important that the windows selected be of the very best quality as regards tightness. This refers to leakage through the frame as well as around the sash. Some double-hung metal sash windows have been particularly bad offenders with respect to infiltration through the frame. In other cases, infiltration through pivoted metal sash windows has been high. The argument is sometimes advanced that it is undesirable to have tightly fitting windows because fresh air is needed for ventilation. The answer is that if ventilation is needed it is better to be able to control the amount rather than to have leakage go on day and night in an uncontrollable fashion even when the building is not in use.

If no ventilating system is installed, windows can be opened slightly if necessary, although even with tight windows there are usually enough air changes except for crowded offices. In some buildings, inferior windows have been installed to reduce the first cost. This is the poorest kind of economy. It results in difficulty in heating, drafts from windows, complaints from building occupants, and high steam consumption.

The windows should always be equipped with the best weatherstripping available. It has been previously stated that the infiltration losses in an office building amount to about 30 per cent of the total losses. This approximation is based on non-weatherstripped metal windows. Weatherstripping the windows probably reduces the infiltration by nearly 60 per cent, thus reducing the total heat loss from the building by over 15 per cent.

Use of Double Windows

The transmission loss through the window glass can be reduced by about 60 per cent by using double windows. In a building where loss through the glass makes up 40 per cent of the total, this means a reduction of nearly 25 per cent in the total heat loss.

The cost of heating the largest office buildings is around $100,000 per year. Anything which will reduce the heat loss from such buildings by 25 per cent is well worth serious consideration. It is believed that double windows will come into more general use in the future as designs are developed which will make the windows easier to clean, this being one of the objections. The increase in the use of air conditioning should also bring about a greater use of double windows, as it is neither necessary nor desirable to have open windows in air conditioned buildings, even in summer.

If rooms are drafty, occupants require a higher temperature than otherwise, thus increasing the cost of heating the building. Windows are one of the principal sources of drafts. The cold down-draft from a window is made up partly by infiltration and partly by cooling of the air in contact with the cold window pane. This draft can usually be eliminated by installing radiators under the windows.

Proper Enclosures for Concealed Radiators Important

In order for the radiators to work effectively, it must be possible for the air in the room to circulate freely through them. If the radiators are to be concealed, the enclosure should be designed to provide free air circulation. A *poorly designed* enclosure reduces the capacity of the radiator and may make it impossible to keep a room warm unless extra radiation is provided to make up for the reduction in capacity.

The air inlet at the floor should be at least 4 inches high and should extend the full length of the radiator. If the warm air outlet is to be located in the front of the enclosure, there should be as much clearance as possible between the top of the radiator and the top of the enclosure. The mistake is frequently made of having the top of the enclosure fit tightly over the top of the radiator so that the air does not have a chance to sweep over the entire radiator surface.

The outlet in the front should reach to the top of the enclosure, and if possible should have a free area at

least 50 per cent greater than that of the air spaces between the radiator sections. It should, of course, extend the full length of the radiator.

Fig. 1 shows enclosures built into the wall and having several characteristics of good design. Note the generously sized inlets at the floor and the outlet grilles having a large percentage of free area. The radiators within the enclosures are of the convection type with their tops even with the bottom of the outlet grilles. One possible objection to the arrangement shown is that the window sills are very wide (about 14 inches), and on cold days the down-draft from the windows appears to attain a sufficient horizontal velocity across the sill outward into the room to cause some discomfort to those sitting in front of windows. It is probable that if the grilles had been placed in the tops of the enclosures, the window draft would have been eliminated.

Preventing Window Drafts with Fan Systems

If a building is heated by a fan system and no radiators are installed, then other means must be used to prevent window drafts if people are to be seated near windows. One way is to use tight-fitting double windows. Another way is to introduce all of the warm air at the windows. Still another way which is being used successfully in an office building is to exhaust all of the air through exhaust openings in the window sills, thus disposing of most of the cold air from the windows before it reaches the floor or has a chance to get out into the room.

The exhaust duct can be located near the outer wall between the floor and the false ceiling of the floor below with short branches leading up to exhaust grilles in the window sills. Fig. 2 shows such an arrangement. An added refinement would be the location of small additional exhaust grilles near the floor in the wall between the windows to catch the cold air which slides down the outer walls.

Store Fixtures Obstruct Radiators

To heat a room properly requires something more than installing the proper number and size of radiators and placing them under the windows. It must be possible for the air in the room to circulate freely to and from the radiators. Any obstruction at the floor which walls off the radiators from the lower part of the room acts as a dam holding back the cold air and preventing it from reaching the radiator in the same way that a dam holds back the water in a stream. The amount of heat delivered to the room is reduced because of the reduced amount of air passing over the radiators. The result is a cold floor and a cold room.

Fig. 3 shows how the circulation of warm air is confined to the upper part of the room; the obstruction of circulation to radiators by store fixtures is very common. In some cases, the cabinets and other woodwork extend solidly from floor to ceiling. In other instances, long show cases with small doors between them block all circulation. The trouble can be easily prevented by providing for an open passage for the air under the fixtures, as shown in Fig. 4, or by leaving a space between fixtures near the radiators. A clear understanding of this problem would eliminate one of the present difficulties in store heating.

Another problem often encountered is that of providing heat for interior offices or spaces in which there are no radiators. Since such spaces have no windows nor outside walls, the amount of heat required is small. The warm air from the radiators travels along the ceiling so if the partition can be left open for a short distance from the ceiling, heat can be introduced in that way. An open space in the partition at the floor allows the cold air to get to the radiators and complete the cycle.

Importance of Proper Doors

On the first floor of buildings, drafts from outside doors are encountered. If all building entrances and first floor shops were equipped with revolving doors, weatherstripped double doors, or storm doors, heating troubles would be greatly reduced. The heat loss from a shop is frequently quite high because of the large area of glass in show windows, the radiators often work at a disadvantage because of being boxed-in or concealed in some way, and then if a large air change is added due to the frequent opening of the outside door, a condition which often results in underheating is obtained. The condition is made worse if the shop occupies a corner and has doors on both sides. One such shop has, in addition, a door leading to the elevator lobby. Whenever the lobby and street doors are opened simultaneously, the air change is very rapid and it is almost impossible to keep the store warm. The use of revolving doors would have almost entirely prevented the trouble.

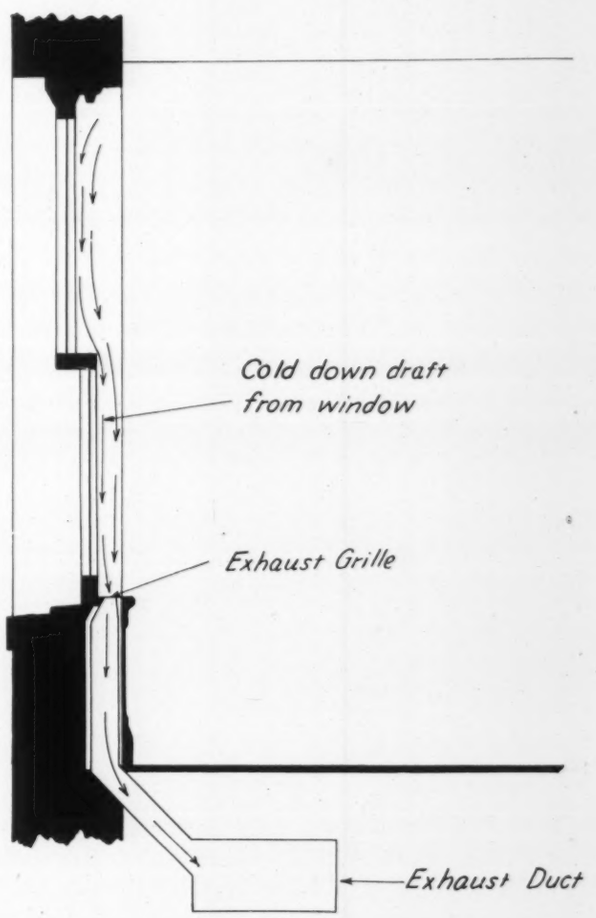


FIG. 2—METHOD OF ELIMINATING WINDOW DRAFT IN A BUILDING HEATED WITH A FAN SYSTEM

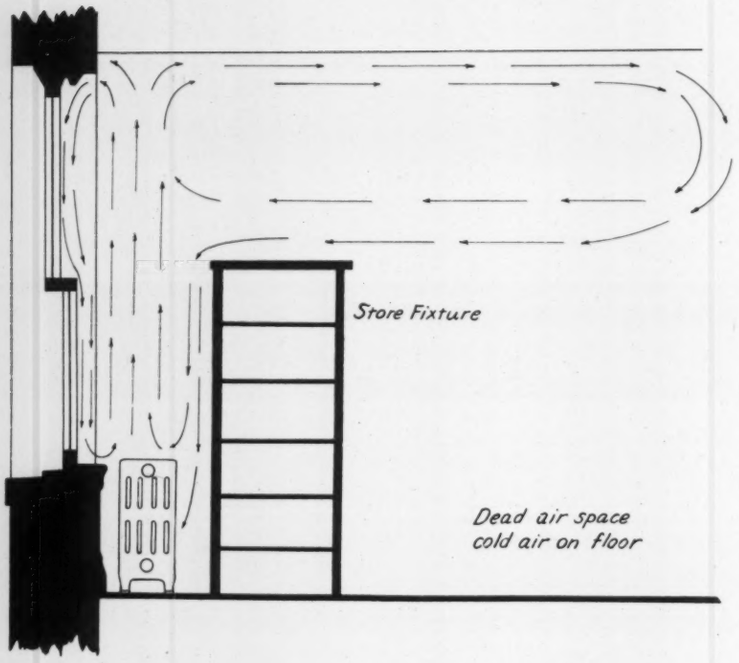


FIG. 3 — CIRCULATION TO RADIATORS OBSTRUCTED BY STORE FIXTURES

Reducing Stack Effect in Tall Buildings

A tall building acts as a chimney. Cold air enters through doors and around windows, mainly in the lower part of the building, and the warm air within the building rises and escapes around the windows in the upper part of the building. Observations made in one tall building indicate that in order to compensate for the greater infiltration on the lower floors and for the rise of heated air to the upper floors, it is necessary to install about 80 per cent more radiation on the 4th floor than on the 40th floor. It is desirable to reduce the stack effect as much as possible because it increases the heat loss from the building. This can be accomplished by tight elevator shafts, tightly fitting elevator doors, and tightly fitting stairway doors.

Water Piping Should Be Protected

An important consideration from the standpoint of heating economy is that it be possible to turn steam off at night when a building is unoccupied. In buildings which do not require heat in the evening, the steam consumption can be reduced almost 50 per cent by turning off steam at night. It is the writer's belief that, if a building is properly constructed and is unoccupied at night, it should never be heated all night except when the temperature is below zero. The building should be de-

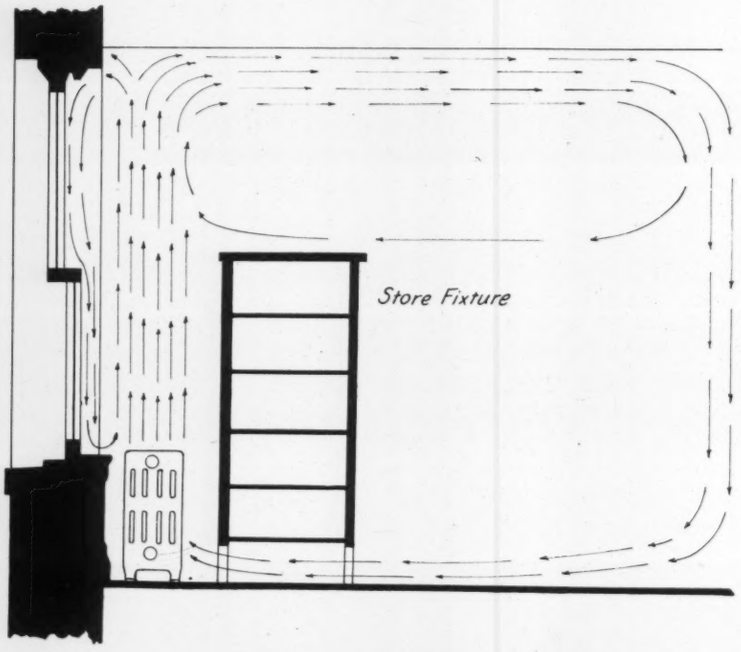


FIG. 4—OPEN PASSAGE UNDER STORE FIXTURES ALLOWS PROPER AIR CIRCULATION

signed with that in mind. Some buildings are heated at night in cold weather and therefore waste steam because of the fear of freezing water piping and causing damage. If water piping is kept out of outside walls and is properly protected, there should be no danger except in sub-zero weather.

Recirculation Saves Heat

It is possible for fan heating to be a wasteful process, the wastefulness depending upon the extent to which warm air is exhausted from the building. Adequate ventilation does not require that all of the air circulated be brought in directly from out-of-doors and then exhausted. It is usually possible to recirculate a large percentage of the air. The heat saved may be 50 per cent or more of the amount required if there were no re-circulation.

It is thus evident that in buildings where there are fan systems, space should be provided for recirculating ducts so that such air as is fit to use over again can be recirculated to whatever extent seems desirable.

FIG. 5—IT MUST BE POSSIBLE FOR THE AIR IN THE ROOM TO CIRCULATE FREELY TO AND FROM THE RADIATORS. ANY OBSTRUCTION AT THE FLOOR ACTS AS A DAM

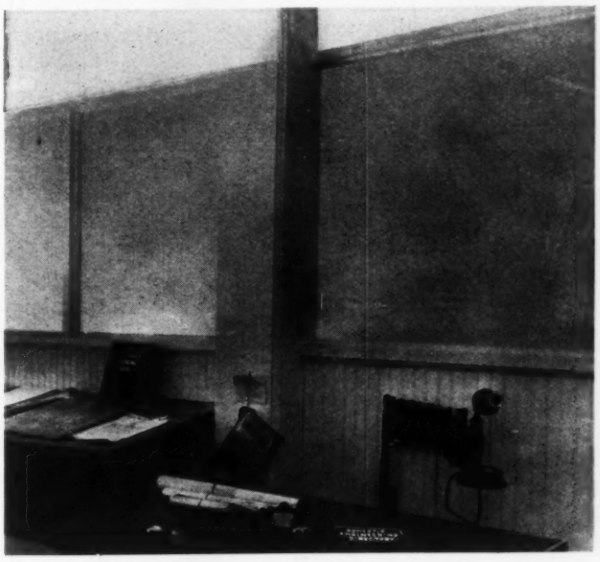


FIG. 6—ANOTHER PROBLEM IS THAT OF SUPPLYING HEAT TO INTERIOR OFFICES IN WHICH THERE ARE NO RADIATORS. IF THE PARTITION CAN BE LEFT OPEN FOR A SHORT DISTANCE FROM THE CEILING, HEAT CAN BE INTRODUCED IN THAT WAY. AN OPEN SPACE IN THE PARTITION AT THE FLOOR ALLOWS THE COLD AIR TO GET TO THE RADIATORS AND COMPLETE THE CYCLE

Importance of the Design

The design of a building should take advantage of every opportunity to make the annual cost as low as is consistent with the use for which the building is intended. Annual cost is intended to mean operating cost plus fixed charges. That is, everything possible should be done to make the cost of operation low provided the investment charges can be justified. The minimum cost of operation is determined by the design of the building. If the operating cost is high because of improper operation or maintenance, the difficulty can always be corrected, but the limitations imposed by the design last throughout the entire life of the building.

Piping *a* Pumping Station

By Orville B. Carlisle* and Duane E. Kennedy†

CHICAGO operates twelve major pumping stations which furnish more than a billion gallons of Lake Michigan water daily. Each of these stations may be considered as a manufacturing plant; the water in the lake is the raw material; and the same water under pressure in the mains is the finished product. An unfailing supply of the finished product is imperative, to which economic production must be subservient. Application of pressure to the water and then the delivery of this water to the mains are the two functions of the pumping station; we shall deal with the latter function in describing the design of the main water piping of the Western avenue pumping station, the most modern in Chicago's water supply system.

Industrial Pumping Plants Comparable to Municipal Plants

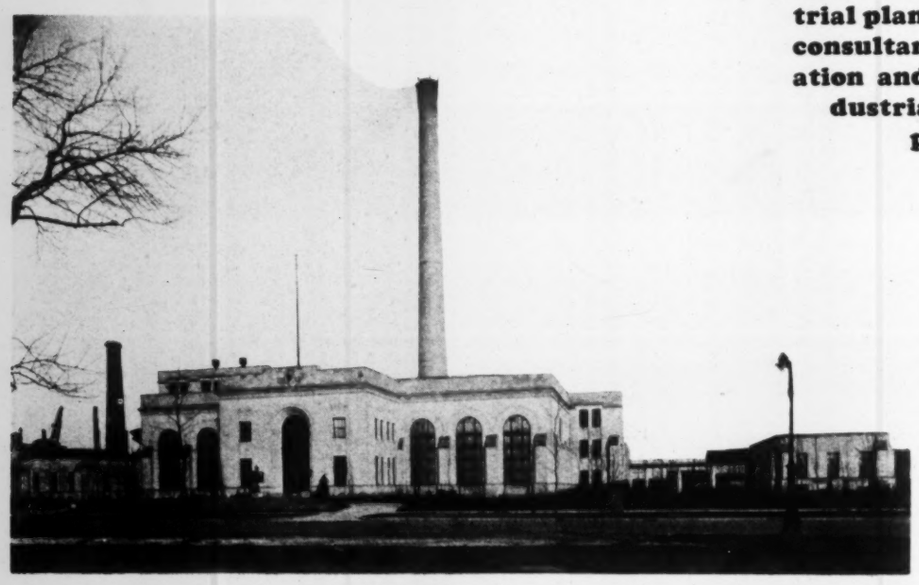
Many industries, to supply water for their own use, require pumping plants, which occasionally are of a size comparable to the average municipal plant. Unlike the condition in the ordinary industrial plant, however, the city of Chicago has had the advantage of experience in designing and operating many pumping stations. The city maintains a division devoted solely to water works design, wherein a progressive policy has brought about many improvements in the more modern pumping stations, showing the result of years of experience from many independent installations. The station at 50th st. and Western ave. offers a water piping design which not only demonstrates the latest practice, but which incorporates many features and details of interest and value to the designer of an industrial pumping plant.

Eleven-Mile Tunnel Through Solid Rock

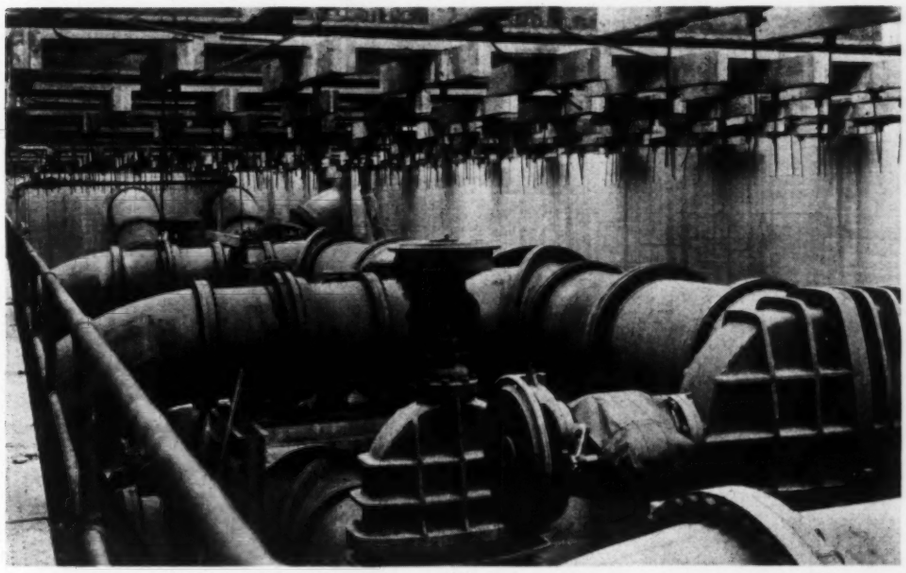
This station, designed under the supervision of Loran D. Gayton, city engineer, is equipped with four centrifugal pumps each rated at seventy-five million gallons per day. They are installed in a pump pit with the center line of the pump shafts eleven feet below datum and about twenty-six feet below grade. Water reaches the pumps by gravity flow, through a concrete lined suction tunnel drilled through eleven miles of solid rock extending from the intake crib in Lake Michigan to the

Many industrial plants operate pumping stations comparable to municipal water works. In describing this Chicago pumping station, the object of this article is to present information which should be of considerable value to the industrial plant engineer and consultant in the operation and design of industrial pumping plants.

* Engineer of Water Works Design, Bureau of Engineering, Dept. of Public Works, Chicago.
† Mechanical Designer, Bureau of Engineering, Dept. of Public Works, Chicago.



EXTERIOR VIEW OF THE WESTERN AVENUE PUMPING STATION



PHOTOGRAPH TAKEN DURING CONSTRUCTION, SHOWING HEADERS AND VALVES IN THE PIPE VAULT

station. Here the tunnel branches at a gate shaft and passes underneath the pump pit, the two branches entering at opposite ends. A concrete bulkhead separates the tunnel branches at the center of the pump room, allowing two pumps to take their suction from the north and two from the south branch tunnels. The water is screened and chlorinated at shafts located at either end of the suction tunnel before its passage through the pumps.

Gate Valve Provided in Suction Piping

Each pump is served by a vertical concrete shaft from the suction tunnel extending up to the lower floor level in the pump pit, from which point cast iron fittings are used to reach the pump. It is noteworthy that the water, in its flow from the lake, enters the first iron piping at this point. The long run from the intake crib to the pumping station has been wholly within concrete lined tunnels and shafts. All suction fittings are forty-two inches in diameter and have flanged connections, with one exception where a bell and spigot joint is used for flexibility. The first of these is specially constructed to form the connection with the top of the concrete shaft. It was set in position and the concrete poured afterward, making a solid connection and an effective seal. As the pumps are set below lake level, it was necessary to provide a gate valve in the suction piping; otherwise, the pump would be flooded at times of low pumpage, and if one pump needed repairing, the other pump on the same half of the suction tunnel would also have to be shut down, thus seriously impairing the output of the station.

Water for Condensers

Water for condenser circulation purposes is taken from a lateral connection on the main pump suction; and, after passing through the circulating pump and surface condenser, is returned through another lateral nearer the pump. This arrangement has proved most satisfactory for supplying condenser circulating water, and is standard when steam turbine driven centrifugal pumps are the main units.

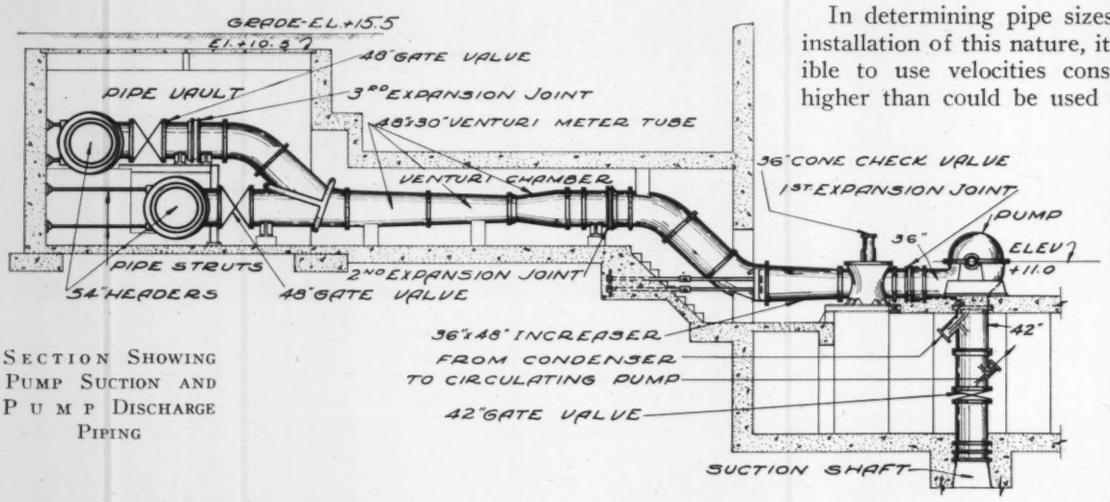
Header System Concentrated in Pipe Vault

Two large distributing mains extend along Western avenue to the north and south from the station, with branches to the east and west. In order to serve these mains from any one of the pumps, and to be able to shut down any particular service without interfering with the others, a rather elaborate header and valving layout were required. It was considered best to concentrate this header system in a room where all valves are accessible from the station. For this purpose a reinforced concrete pipe vault was built in front of the station having its top just below grade. It is connected with the main pump room by four tunnels in which are located venturi meter tubes for measuring the flow from each pump. By climbing a short flight of stairs from the pump room, the station operators may enter any venturi tunnel leading to the main pipe vault. The entire top of the pipe vault is constructed of removable concrete slabs supported upon steel beams. Should any fitting or valve need to be replaced, it may be lowered into position from a point directly above without disturbing the adjoining fittings. The vault is heavily waterproofed, and the top covered with soil and sod, giving no outward indication of its presence.

Sectionalizing and Valving Headers

The accompanying diagram shows the general layout and valving of the discharge system. Water leaving any one of the main pumps first passes through an automatic

check valve of a new design, continues on through the venturi meter tube, and then through a lateral fitting to the two main water headers. A gate valve is placed adjacent to each header in the pipe lead from the pump; thus any pump may discharge to either or both headers independently of the others. The headers are divided into two sections by valves at the center, two pumps being connected to each section. The ends of the headers are also valved before their juncture with the street mains, and cross-connections between the headers are provided at either end of the vault. With this arrangement it is obvious that either header may be segregated, thus maintaining service to all of the street mains; moreover, a single pump may be employed to supply a particular service main. The latter feature permits measurement of the flow in one service main without interfering with the normal operation of the others.



SECTION SHOWING PUMP SUCTION AND PUMP DISCHARGE PIPING

Increasing Pressure on One Main

Occasionally a condition exists which makes it desirable to increase the pressure on a certain service main while maintaining normal pressure on the others. This may be accomplished easily by increasing the speed of one pump and segregating it to the service main in which the additional pressure is desired.

An incident occurring a short time ago demonstrated the effectiveness of this plan of operation. At that time it became necessary to close down completely another pumping station while alterations were made in the suction well. This station is located in the extreme southern part of the city, and entirely apart from the district normally served by the Western avenue station. By the plan described, however, it was possible to maintain sufficient pressure in the other district through the medium of the Western avenue station so that not a single complaint was received during the time required for the alterations. The remote parts of the other area received water at about twenty pounds pressure, a sacrifice of only a few pounds from normal conditions.

Control of Motor-Operated Valves

Sixteen gate valves are required to control the discharge system. All are of the solid wedge disc type with outside screw and yoke. They are operated by waterproof electric motors acting through cut-tooth gearing. The mechanism embodies an automatic mechanical hammer-blow device which permits the motor to come up to speed before picking up its load. This feature is necessary to dislodge the valve disc from its seat; otherwise the motor could not develop sufficient torque.

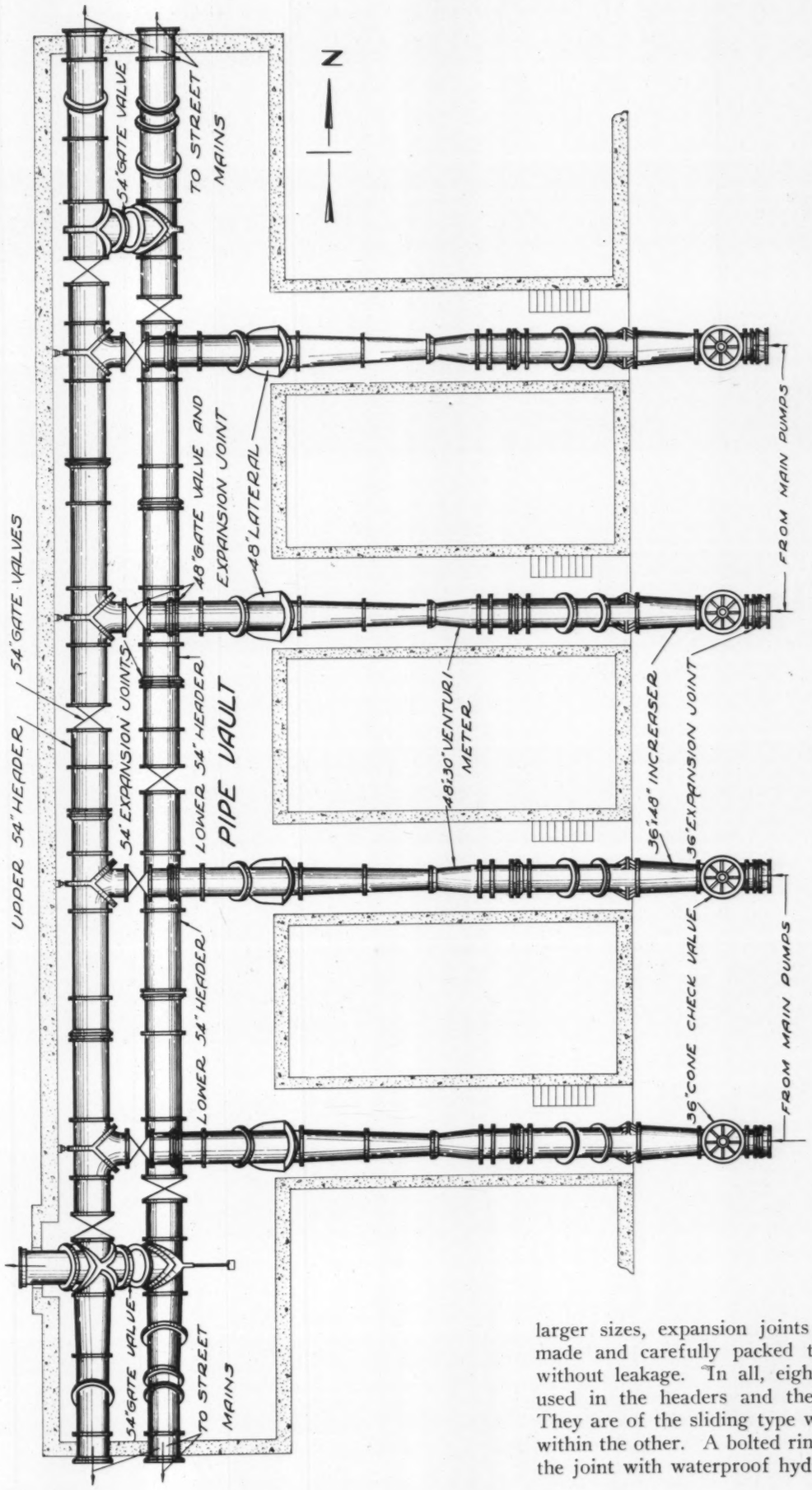
Each motor may be controlled from either of two control stations, one of which is in the vault near the motor and the other in a central control room in the office section of the plant. Here, sixteen panels are grouped in the form of a square, having four panels to a side and space for entrance at the corners. Within the square the operator may observe the position of the valves, and may actuate any of them without moving more than a few steps. Colored lights on the control panels indicate at a glance the open or closed position of each valve.

Proper Velocities of Flow

In determining pipe sizes for an installation of this nature, it is feasible to use velocities considerably higher than could be used through the long distributing system. Each pump has a 36-in. diameter discharge connection; which, with a rated output of 75,000,000 gal. per day, gives a velocity of 16.4 ft. per sec. This size is maintained through the check valve where the use of the smaller sizes is most advantageous. An increaser immediately follows the check valve, however, enlarging the pipe to 48-in. diameter, and lowering the velocity to 9.24 ft. per sec. for the same pumpage. This is still excessive for a long run, but is preferable for use through the venturi meter, insomuch as the velocity through the throat of the meter must be high at maximum output in order to give a sufficiently accurate reading at low pumpages. Thus, the meter tube may be of reasonable length and yet possess a gentle convergence and divergence of the stream in passing through the throat. The headers in the pipe vault and their cross connections are built up entirely of 54-in. diameter material; which, with the divided flow occurring in supplying the several services, reduces the velocity until the friction pressure drop diminishes to an amount satisfactory for economical distribution.

Use of Expansion Joints

The normal pressure curve at this station demands discharge heads as high as 150 feet. Within the station it was necessary to use fittings of special design. Flanged joints were adopted. Expansion joints are required where flanged material is used, offsetting to a certain extent the advantages gained by the flanges. In the



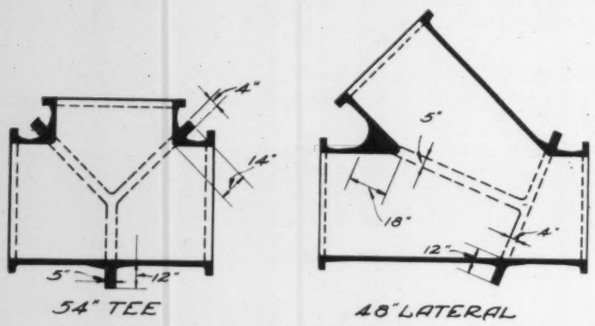
PLAN VIEW OF PIPING IN VENTURI CHAMBERS AND PIPE VAULT

larger sizes, expansion joints must be very accurately made and carefully packed to provide for expansion without leakage. In all, eighteen expansion joints are used in the headers and the pump discharge piping. They are of the sliding type with two parts, one sliding within the other. A bolted ring follower is used, sealing the joint with waterproof hydraulic packing. Both sec-

tions of the joint and the ring follower are lined with ⅛-in. finished copper, upon the surface of which all movement takes place.

Thorough Anchorage Important

The use of expansion joints makes thorough anchorage of the intervening pipe and fittings of greater importance than would be necessary otherwise. For example, consider the discharge lead from one of the pumps. Here it is essential not only to provide for expansion in the pipe, but also to prevent stress on the pump, and to guard against thrust due to water flow within the pipe. Starting at the pump, the first expansion joint bolted to the pump discharge flange removes the possibility of transferring stress from the pipe to the pump casing. The next section includes the check value, increaser, and several fittings which form an offset. The elevation of the pipe is raised nine feet by the offset, bringing it to the level of the venturi meter



Left—54-In. Tee. Right—48-In. Lateral

tubes, where the second expansion joint is bolted to the upper elbow fitting. Special anchorage is required in this section due to the change in elevation. It has been secured by four rods of 2½-in. diameter embedded in a heavy reinforced concrete block, the ends of the rods projecting through heavy lugs cast integral with the lower elbow fitting. Turnbuckles at the mid-points of the rods serve as adjustments for bringing the fittings into proper position. A concrete pier is cast beneath the lower elbow fitting and another above the upper elbow fitting which bears against the roof of the venturi chamber. The check valve has its own structural steel support.

Following the pipe line from the second expansion joint, the next section includes the venturi meter tube, the lateral fitting, gate valve, and tee in the lower header, together with a 45° fitting in the lateral branch to the upper header. For this section, the tee in the lower header serves as a fixed anchorage point, being supported horizontally from the wall of the pipe vault. Expansion in this section is provided for at the second expansion joint, the lateral carrying with it the 45° fitting leading to the upper header.

The last section, beyond the third expansion joint, includes only the gate valve and tee in the upper header which is anchored in the same manner. The horizontal support for both tees consists of a pair of 4½-in. pipes, bolted to the tees and attached to the wall through bearing plates. It is well to note the amount of unbalanced thrust which must be resisted by these horizontal pipe struts. For 48-in. diameter tee outlets, and 60 lb. per sq. in. working pressure, this represents a total force of over fifty tons.

In the main headers, expansion and anchorage are provided for in much the same manner as in the pump discharge piping. Here base fittings are utilized as anchorage points at the centers and ends of the headers. Expansion joints are placed in the three spaces between the pump connections.

Designing for Pressure Due to Water Hammer

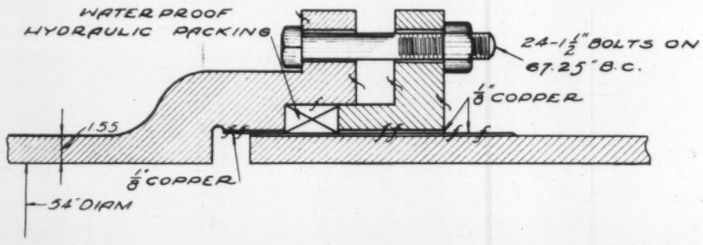
Special design of tees, crosses, and laterals was necessary for this installation, stresses being based on a hydraulic pressure of 186 lb. per sq. in. This pressure represents a momentary condition only, which may occur from water hammer, whereas the normal maximum pressure is about 65 lb. per sq. in. The actual amount of stress developed from water hammer is uncertain and difficult to determine theoretically. It was therefore decided to design the fittings for an arbitrary pressure of 100 lb. per sq. in. more than the designed working pressure for class B cast iron pipe, which is 86 lb. per sq. in.

Reinforcing Large Fittings

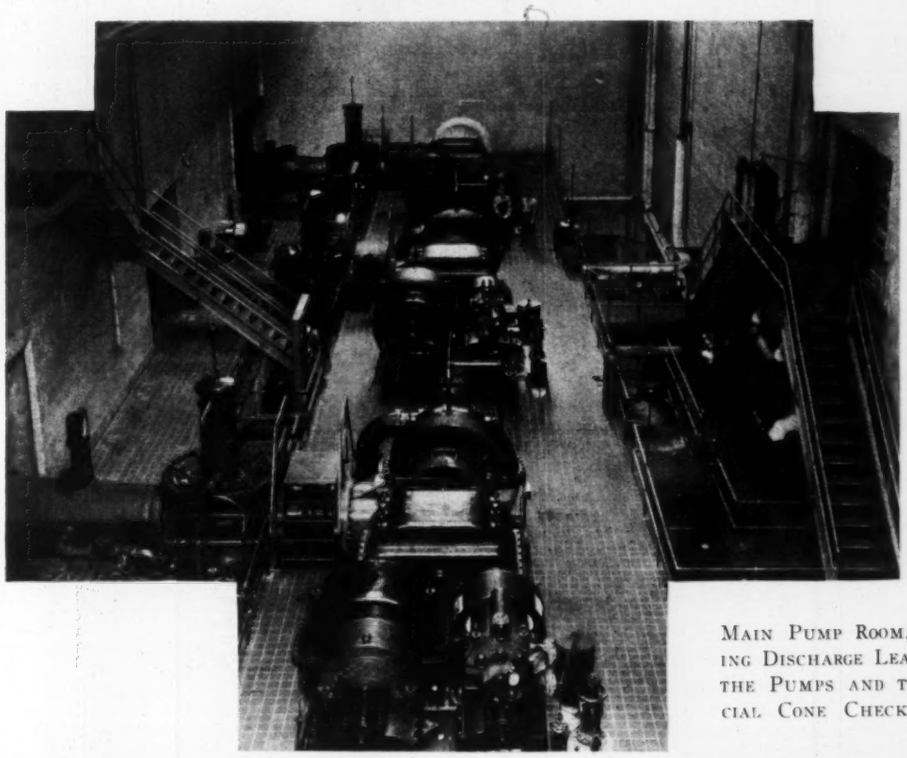
Fittings were designed with external ribs, not subject to the erosion and corrosion acting upon internal stiffeners. In determining the rib sizes, the walls of the fittings were assumed to take ring tension stresses only. Castings of 28,000 lb. per sq. in. ultimate fiber strength were specified and in order to comply with this semi-steel castings were furnished. Since the maximum fiber stress is only 4,400 lb. per sq. in. under normal operating conditions, there exists a factor of safety of 6.3 to withstand shock or water hammer. In casting the fittings of this design, great care was used in cooling, to prevent internal stresses being set up in the metal, inasmuch as the ribs are much heavier than the walls.

Metering the Output

The venturi meters are of standard design, and as previously mentioned, measure the amount of flow from each of the main pumps. They have 30-in. diameter throats, through which the water must pass after constriction from the 48-in. entrance diameter. A combined indicator, recorder, and integrator, provided with each meter tube, is placed in the pump room within sight of



Fifty-Four Inch Expansion Joint



MAIN PUMP ROOM, SHOWING DISCHARGE LEADS FROM THE PUMPS AND THE SPECIAL CONE CHECK VALVE

the operating engineer. These instruments—together with water pressure gages—furnish a constant record of the output of the station in terms of million foot-gallons. This is a practical unit of output and serves as a base for determining all unit pumping costs. At this station, the venturi meters perform a second function, which is in some respects of even greater importance than the measurement of flow—namely, the controlling of the chlorine machines, which are connected to the venturi tubes and which automatically add a chlorinated water solution in direct proportion to the amount of pumpage.

Check Valves in Discharge of Pumps

Customary practice demands that a check valve be placed at the discharge of a centrifugal pump to prevent back flow, should the pump, for any cause, stop or slow down. A simple swing check valve is usually satisfactory for a small pump, or for a moderate sized pump if the head is low. However, where the pipe size is large and the head relatively high, a swing check valve often receives a terrific shock upon reversal of flow.

The check valves used at this station are of a new design. Before their adoption they were thoroughly tested at another station, and proved successful under severe usage. They are of the tapered plug or cone type, having a valve disc of this shape, fitting a conical valve body. The waterway is opened or closed by turning the plug 90°; the first position presenting a clear opening of full pipe size, and the second position a complete closure of the port. The valve is entirely automatic, not only closing the port automatically upon decrease of pump pressure, but opening the port also when the pump has built up a pressure exceeding that on that downstream side of the valve.

Power for operating the valve is furnished by a hydraulic piston traveling in a cylinder above the valve body. Water pressure for use in the hydraulic cylinder is obtained either from the pump side of the valve or from the downstream side, depending relatively upon the closed or open position of the valve. The valve may be made to function manually if desired, but normally it is allowed to act automatically, responding to small differences in pressure up and downstream of the valve. When a pump is to be started and put into service, the valve remains closed until the pump reaches a speed such that the discharge pressure exceeds the pressure beyond the check valve. Only a small difference in pressure is required to operate the control valve, which in turn opens the valves to the hydraulic cylinder. The opening of the valve is thus accomplished and the pump thrown into service. The opposite takes place when, with a reduction of speed, the pump loses head until it becomes less than that in the service headers. It is important that any check valve used for this service shall act promptly, before the reversal of flow has time to gain appreciable velocity. This means that the entire operation must consume but a few seconds.

Several pumping stations operated by the city have pumps which are directly driven by electric motors. With this type of pump drive, reversal of flow takes place even more quickly than with the steam-turbine-driven unit. The first reason for this is that the most frequent cause of an emergency shut-down is the failure of electrical supply to the motor, resulting in instantaneous cessation of driving power. The steam-turbine-driven unit, in comparison, is shut down by the hand throttle for most troubles that necessitate its removal from service. This requires some time, allowing the pump impeller to gradually diminish the force exerted upon the water. Even though the steam is suddenly shut off by operating the emergency trip on the throttle valve, the vacuum upon the lower stages is not lost at once. The second reason for a slower reversal of the

pump with steam turbine drive, is the greater inertia of the moving parts, tending to keep the pump in forward motion.

Insulation of Pipe Eliminates Electrical Troubles

One must not overlook the effects caused by condensation where large water mains are exposed to moisture laden air. There is an enormous area of pipe surface, every part of which serves as an effective condenser. Considerable expense was incurred to properly cover the entire system which is open to the surrounding air in the venturi chambers and pipe vault. Unfortunately, it was necessary to start service from the station before the covering was applied; this, however, afforded an opportunity to compare the conditions with and without pipe covering. Operating with bare pipe, sweating was so severe that the pipe vault was continuously wet. The excessive moisture tended to cause short circuiting of the electrical relay wiring, requiring constant attention to keep the valves in an operative condition. Upon the application of the covering, the vault and chambers became dry, providing a clean and comfortable place for those who were required to work there and putting an end to the electrical troubles which had occurred previously. Wool felt insulation with a sewed jacket of heavy canvas is used for the pipe covering. The whole is finished by painting, so that the installation presents a neat and finished appearance.

Care Exercised in Installation of Piping

Wherever heavy flanged piping is used, too much emphasis can not be placed on the need of accurate workmanship. This applies not only to shop work, but also to the field installation. In the first place, flanges must be cut to accurate face-to-face dimensions, and the faces must be absolutely true. It is impossible to draw up a joint of this size unless it fits beforehand. For this installation, all flanges were finished 1/32 in. short to allow for gaskets, and seldom has there been a large flanged pipe job which has been completed with as little trouble as the one described. It would not be unreasonable to say that everything fitted perfectly.

It was necessary that the field installation work be done accurately and carefully. Fittings had such great weight that they could not be allowed to strike against each other or against any solid object without endangering the flanges. Each piece was securely held after it has been brought into position, so that the bolts could be drawn up to distribute pressure evenly upon the gasket. The two sections of the expansion joints were accurately aligned, and were perfectly concentric before the packing and gland follower were applied. Great care was necessary to compress the packing uniformly without over-tightening some of the follower bolts. This task appears more formidable if one considers the circular packed joint by its lineal dimension, which is more than fifteen feet in the case of the 54-in. diameter.

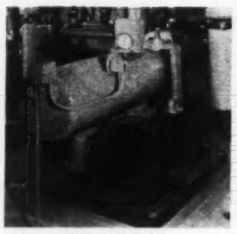
Generally one length of pipe or a fitting is left as a closing piece, the dimensions of which are not taken until all others are in place. This is especially necessary where offsets and angle fittings are used. Wherever possible, a closing piece is selected which may be set into place without disturbing the adjacent pieces, such as an elbow fitting or a piece next to an expansion joint. In the pump discharge leads to the headers, the upper elbow fitting in the offset was used for this purpose, its dimensions being taken after the other fittings were erected.

Actual operation of this station has shown that this piping layout has satisfied the needs for flexibility, for ease in operation, and for accessibility and the extra investment required to obtain these results was well justified.

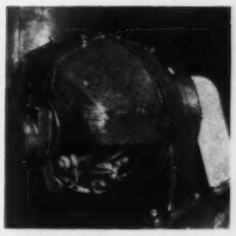
Piping a Power Plant Near the Arctic Circle to Be Described

A complete modern power plant is being erected to supply electrical energy and steam for the eight million bushel grain elevator on the west coast of Hudson's Bay, almost within the fringe of the Arctic Circle. Because of the isolated location—and because a shut down during the wheat shipping period would be extremely costly—the plant is designed for absolute reliability. For the present, it is expected that the plant will operate during about three months a year. R. A. Hanright describes the interesting piping features of this unusual plant next month.

The control of heat, of sound and of light is the subject of Edgar C. Rack's paper in the September HEATING, PIPING AND AIR CONDITIONING. While the heating and air conditioning engineer is interested primarily in the utilization of modern building insulation for the first-named purpose, he is also concerned with the other two factors. It is felt that this informative article will be of considerable value to such engineers.



Engineers in charge of air conditioning and heating equipment often are inclined to neglect the proper care of the auxiliary electric equipment used for driving fans, pumps, etc. In this article, the author tells how operating expenses can be cut down by the maintenance of proper voltages. Just how over or under voltages increase costs is described in detail, and the remedies are pointed out.



Saving Money in Motor Operation

By Louis Mackler*

THERE are certain fundamental factors that play a prominent part in the successful performance of electrical equipment. A knowledge of those factors will be found of value to those who are associated with the different branches of engineering in which electricity is used. The air conditioning and heating engineer frequently has a certain amount of electrical equipment under his supervision, and must be familiar to some extent with the fundamentals of its operation.

Efficient operation of electrical apparatus depends largely on the maintenance of proper voltage values at the terminals of the apparatus and keeping this voltage free from sudden increases or decreases. The maintenance of a proper and steady voltage is in turn dependent upon the type of the electrical distribution system that supplies the current, the general nature of the connected load and the characteristics of the feeder system employed.

The Distribution System

For direct current distribution the method in extensive use today is that known as the *Edison 3-wire system.* This provides a potential of approximately 115 volts between the neutral, and either the positive or negative side, and 230 volts between the positive and negative feeders. Motors of 5 hp and larger are generally connected to the 230 volt mains; lamps are invariably operated on the 115 volt feeders. Due to the comparatively low voltages this system cannot be employed for long distance distribution.

On the other hand, alternating current, with the use of transformers, may have its voltage raised and lowered at will. The methods employed to distribute alternating current are many and varied and combinations of phases and voltages may be arranged to suit almost any condition.

Voltage fluctuations are more common on alternating current systems than on direct current. The reason is that, generally, the direct current distribution feeders are connected into a complete "grid" or "network" thus making the entire capacity of the system available to take up sudden load peaks. In alternating current installations, each plant or building, or perhaps several plants or buildings, is supplied from an individual transformer. The result is that sudden peak loads have to be absorbed by the capacity of the transformer which may produce a more or less reduced voltage condition. With the introduction of the recently perfected alternating current network system this condition will be somewhat remedied.

Before deciding upon the type and character of the electrical apparatus for any specific installation it is very important to ascertain the nature of the electric current supplied at the particular location in question. This information may be obtained from the proper officials of the local electric light and power company. In making this request it is also wise to indicate the expected time of completion of the project as, due to the rapid progress that is taking place in methods of electrical distribution, six months or a year may see a totally different system in use.

The nature of the connected load in a plant will have great bearing upon the steadiness of the voltage in the feeder system. Direct current motors started through resistance grids or similar devices and thus limiting the initial inrush of current produce little disturbance; induction motors on the other hand, will, regardless of the method of starting, produce a greater amount of voltage fluctuation. If heating devices are employed, efforts should be made to have the load balanced as equally as possible across the various phases and conductors. This will tend to lessen the voltage fluctuations when large increments of load are connected to the line. As a general thing, however, electrical equipment in a plant is a fixed quantity and steps to relieve low voltage conditions and fluctuations must of necessity be concentrated on the causes that resist current flow and thus tend to interfere with the maintenance of proper voltage values.

Resistance to Current Flow

Water flows through a pipe because a head or pressure exists, the rate of flow depending directly upon the amount of the pressure and inversely upon the resistance presented by the pipe. In a similar manner the flow of an electric current is governed by the pressure or voltage available and the total resistance of the complete

* Consulting engineer, New York City.

electric circuit including the conductors, switches, fuses and other devices and of the actual equipment to which current is supplied. This resistance is present in both direct and alternating current circuits and apparatus.

In addition, the flow in an alternating current circuit is also retarded by two other factors which are of considerable importance—*reactance* and *skin effect*. The latter phenomenon may be compared to the centrifugal action that occasionally occurs in the flow of a liquid in a pipe. This action forces the liquid to the outer sections of the inside of the pipe leaving the central portion vacant. Skin effect becomes increasingly evident in conductors that are above 400,000 circular mils in area.

Reactance may be compared to the back pressure that exists in ventilating duct systems and is caused by the magnetizing current in motors, transformers and other apparatus containing coils of wire. There is also a certain amount of reactance present in the feeders themselves, particularly if they are spaced some distance apart as may be the case when installed on racks or hangers in a factory or other industrial establishment.

These three vital factors—resistance, reactance and skin effect—have great bearing upon the successful maintenance of proper voltage values. A most important fact is that the detrimental effects they produce may be materially decreased with the expenditure of a reasonable amount of money.

Effect Upon Heating Devices

Heating by electricity is usually accomplished by sending a current through a coil or other conducting medium that generally possesses a fixed resistance. The rate at which heat is developed is directly proportional to the square of the current or

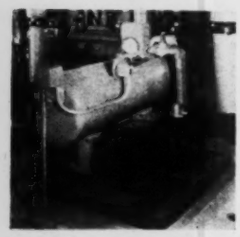
$$\text{Btu per hour} = 3.413\ RI^2$$

where R is the resistance of the medium in ohms and I is the current in amperes.

But the current that flows is dependent upon the voltage at the terminals of the apparatus and the voltage in turn is dependent upon the total resistance (plus reactance and skin effect in alternating current systems) in the circuit including that of the conductors, switches, fuses, etc.

Assume a heating device designed with a resistance of 11.5 ohms (R) for operation on 115 volts (E). The current (I) will then be 115 divided by 11.5 or 10 amperes, as $I = E/R$. Further assume that due to the inadequacy of the connecting feeders, the voltage at the terminals has a value of only 105. The current therefore decreases to 9.1 amperes. At normal voltage the heat is

$$\text{Btu per hour} = 3.413 \times 11.5 \times 10 \times 10 = 3924.9$$



In the second case, with a reduced voltage of 105, the current would be 9.1 amperes and the heat is

$$\text{Btu per hour} = 3.413 \times 11.5 \times 9.1 \times 9.1 = 3250.2$$

This is a reduction of 674.7 Btu per hour or 17.1 per cent. In other words with a drop in voltage of a little over 8 per cent the heating output is decreased over 17 per cent.

One advantage of heating by electricity is the accuracy of control possible—yet such a valuable feature may be totally nullified by reduced or continually fluctuating voltages. In those applications where quantity production is involved, such as the baking-on of enamel or japan finishes on completed products, a material decrease in heat will result in lessened output or in defective workmanship. Such conditions justify a considerable expenditure to provide the necessary remedies.

The tendency to employ electricity in hot water storage systems brings about a condition where it is essential that the required heat be produced within a more or less definite time interval. The control devices used with these systems provide for voltage irregularities and for the increased time necessary to heat a definite quantity of water to a pre-determined temperature but the efficiency and economy are enhanced by the application of a current having the proper voltage.

Effect Upon Motor Applications

The manufacturers of electric motors will generally guarantee that "the motors shall operate successfully at normal rated load at any voltage not more than 10 per cent above or below the name plate rating, but not necessarily in accordance with the standard of performance established for operation at normal rating." As a result of improvements in design there has also recently come into existence a *service factor* of 1.15 which states that "when authorized by the manufacturer, motors will carry continuously 1.15 times their rated load when the voltage does not vary more than 10 per cent or the frequency more than 5 per cent, or the combination of voltage and frequency does not vary more than 10 per cent above or below normal."

In addition to the above, motors are also usually required to not exceed a temperature rise of 40 degrees Centigrade above the surrounding air when delivering their rated capacity.

These limitations constitute the fundamental bases for motor design, application and performance and attention is specifically directed to the fact that, in two of the three requirements, the maintenance of proper voltage is a paramount consideration. It will be shown presently that the third condition—temperature rise—is also directly dependent upon proper voltage, omitting discussion at this time of the effect of overload and of frequency fluctuations.

As a general proposition the fluctuation of voltage on a motor circuit is the equivalent of starting it too rapidly, producing a heavy mechanical shock and almost a hammer-blow effect, depending upon the extent of the voltage surge. If the motor is connected to apparatus having a comparatively large inertia a backlash action may result due to the abrupt change in speed of the motor. Thus for example, a housed centrifugal fan has a relatively larger diameter and a higher speed of rotation as compared with a centrifugal pump of the same horsepower. Therefore, the motor of the former device would be subjected to severe mechanical and electrical strain in attempting to regain normal speed after a sudden voltage drop.

If the voltage of the feeders, because of the resistance, reactance and skin effect factors previously discussed,

is continually below the normal rating of the motor, a reduction in the speed, particularly of the direct current type, will result, the heating will be increased, tending to limit the available capacity, the efficiency will be lowered and the starting and the pull-out torque will be decreased. About the only benefit that occurs due to a lower voltage is in the case of the alternating current induction type motor where a slight increase in power factor takes place. This, however, is overbalanced by the concurrent detrimental conditions.

Unbalanced Voltages Across Phases: On 3-phase alternating current circuits another serious situation arises where the voltages across the phases are unbalanced. This unbalance may be brought about by the single phase load on the feeders being arranged in a non-uniform manner. Such is particularly apt to be the case on a 3-phase, 4-wire, 120-208 volt network system where a lighting load may be imposed on one or two phases of the 120-volt conductors, leaving the other phase or phases with a resulting higher voltage.

Test on a 15 hp Motor: Where direct current is employed, shunt wound motors are often used for the operation of centrifugal pumps, fans and similar devices. The table below lists the conditions that arose with a 15 hp motor supplied with current at different voltages. Throughout the test the horsepower output was kept constant except when obtaining the speed at *no load*.

Voltage—Percent	Speed—No Load	Speed—Full Load	Efficiency
120	1450	1320	84.7
110	1395	1240	85.2
100	1350	1150	85.3
90	1315	1050	84.8
80	1300	935	83.5

It will be noted from this table that the efficiency was highest at normal voltage and that increase of 20 per cent in the voltage reduced the efficiency 0.5 per cent. This was due to the increase in heating because of the greater current flowing. On the other hand reduced voltages decrease the speed, thereby lessening the ventilating effect in the armature. A 10 per cent drop in voltage produces an approximate decrease of from 6 to 8 per cent in speed. Such a decrease is not easily detected and, in certain instances, may result in lessened output without the cause being immediately recognized.

Effect on General Purpose Induction Motors: On alternating current applications the results are somewhat more complicated. In the case of general purpose induction motors, increasing or decreasing the voltage has the effect of changing the rating with regard to power factor and efficiency in proportion to the square of the voltage. For example, a 10 hp motor supplied with a voltage 10 per cent above normal would have the characteristics of a 12 hp motor, yet the heating effect would be greatly increased—probably beyond the limiting requirement of a 40-deg. Centigrade rise above the surrounding air.

Speed of A. C. Motor: The speed of an alternating current motor is a direct function of the frequency, allowing for a certain percentage of slip or lag. This figure is generally about 5 per cent with normal voltage. An increase of 10 per cent in the voltage will reduce this slip about 17 per cent, thus causing a corresponding increase in speed, which under certain conditions may be undesirable; a voltage decrease of 10 per cent will result in an increase in the slip of about 21 per cent, causing a corresponding decrease in speed which in the vast majority of cases is detrimental to the operated apparatus.

On the other hand either increases or decreases in applied voltages may generally cause greater heating effect than when the motor is operated at normal voltage—that is, the heating effect is least under the latter condition. Motors are occasionally installed in exceedingly warm places; this makes it incumbent that they shall not be overloaded and that they shall be supplied with the proper voltage.

Results of Unbalanced Voltages: The worst condition that 3-phase alternating current motors may be subjected to is when they are supplied with unequal or unbalanced voltages. The causes of such unbalancing have been previously explained. If the voltages differ by 10 to 15 per cent the current flow in the different portions of the windings may vary by 30 to 50 per cent above the normal balanced status. This produces a greatly increased heating effect, often exceeding the specified or allowable safe limits and in extreme cases necessitating the installation of motors having an excess horsepower rating of from 15 to 25 per cent above the calculated size. Both the efficiency and the power factor are greatly reduced and if a plant is operating under a current contract with the electric light and power company that contains a penalty power factor clause, this unbalanced operation may cost considerable sums of money.

The table below shows the temperature results of a test made on a 35 horsepower, 3 phase, 25 cycle, 2 pole motor under the designed balanced voltage of 173 and under unbalanced voltages of 167-145 and 165.

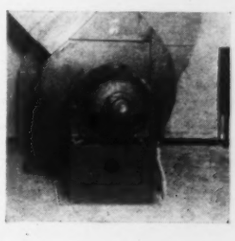
Volts Each Phase			Amperes Each Phase			Temperature Rise—Degrees Cent.				
						Prim. Core	Primary Coils			Rotor
							A	B	C	
172	173	172	105	113	115	37	37	35	32	30
167	145	165	155	124	69	48	66	51	41	60

It will be seen that certain parts of the motor have temperatures considerably in excess of the 40-deg. rise.

The following table indicates the drop in efficiency and power factor that existed in the case of a 30 hp, 60 cycle, 3 phase, 8 pole motor operating under a balanced 440-volt condition and under unbalanced voltages of 446-380 and 430.

Voltage Each Phase			Amperes Each Phase			Efficiency	Power Factor
440	440	440	40.5	40.5	40.5	88	87
446	380	430	53	52	24	83.5	84.7

An unbalanced difference of 6, 60 and 10 volts across the 3 phases resulted in lowering the efficiency and power factor 4.5 per cent and 2.3 per cent respectively, a fairly high price to pay for such a condition.



Mention was previously made

of a service factor of 1.15 wherein a manufacturer will allow this amount of overloading above the normal rating provided the voltage does not vary more than 10 per cent up or down. This is a valuable feature as it leads to a reduced initial investment, lessened operating charges and decreased interest on plant. For instance, suppose a fan or a pump requires 11 horsepower; by being in a position to take advantage of this service factor a 10 hp motor would be adequate.

Remedies

The electrical portions of boiler and engine rooms, industrial plants and similar establishments generally receive too little attention and are expected to operate at high efficiency. The maintenance of proper voltage values is a keystone in the arch of successful operation of electrical apparatus. A thorough examination of the electrical equipment and the calling in of expert help, if necessary, will pay immediate dividends.

A voltmeter should form part of every switchboard. In addition portable voltmeters should be kept available. Where load has been added on existing feeders a check should be made to see to what extent the voltage has been reduced at full load operation.

Where alternating current feeders are installed exposed on racks or brackets, and it is possible to cut the distance between the feeders to two-thirds or one-half, it should be done. This will assist in decreasing reactance.

If the voltage drop is excessive along certain feeders, additional feeders should be installed in multiple or the load more equally distributed. The engineer in charge should have a record indicating the voltage available at every item of electrical equipment and this should be checked at regular intervals to insure proper working conditions.

If the voltage at the point of service entrance connection is more than 5 per cent below normal when the plant is operating at about 60 to 75 per cent of full load, the condition should be called to the attention of the local electric light and power company for rectification.

It is important for the man in responsible charge of the operation and maintenance of electrical equipment used in connection with air conditioning and heating systems to institute a system of regular voltage readings and recordings; these duties and records should be given the same care and consideration as is supplied to similar data pertaining to fuel consumption, steam or water pressure maintenance and the many other items that enter into the modern plant or industrial establishment.

Warm Air System Conditions Lodge Hall

By Platte Overton*

THE benefits of some measure of air conditioning for practically every type of building are recognized widely. A modern direct-fired, hot blast (usually called warm air) system may be used with satisfaction to heat, humidify, cool and clean the air for lodge halls and public buildings of that type. This has been demonstrated by the installation of such a system in the Masonic Temple at Burlington, Wis.

The building is 80 ft. square and is 32 ft. from the ground floor to the ceiling in the lodge hall. Twenty-two rooms are heated and air conditioned. Part of the building is arranged to be rented to the public for plays, dances, and similar entertainments.

Problem to Revise Original Plans

As the original plans for the lodge hall provided only for heating by exposed radiation, the installation of the necessary fans, washers, supply and return ducts offered a real problem. Ducts were left exposed in the kitchen, but in other parts of the building they were concealed in partitions and attic spaces. Supply ducts in the attic spaces are uninsulated.

Oil-Fired Heater Used

The system consists of a large oil-fired heater, together with the necessary fan, air washer and the system of automatic temperature control. The heater is of standard make, is constructed of cast iron, and weighs 8,500 lb. At a 12-lb. combustion rate and 68 per cent efficiency it will give 12,000 c.f.m. of air a 100-F temperature rise or 20,000 c.f.m. a 60 F rise. Eighty per cent efficiency was obtained in a test, using oil as a fuel. Velocity of the air over the heater casings is 1,700 f.p.m.

The oil burner is operated by a furnace control in the warm air plenum, which starts the burner at 80 F and stops it at 180 F. In zero weather, the burner runs for about 5 minutes and remains idle for 15.

The fan is propelled by a variable speed motor; positive cut off dampers are placed in all warm air ducts so that any part of the system may be shut off. Control is by room thermostats; in cold weather, no part of the building is allowed to cool below 50 F.

Air Recirculated

Ninety per cent of the air is recirculated, reducing the fuel bill materially; only enough air to compensate for that exhausted from the kitchen and toilets is supplied from the outside. In mild weather, however, outside air is also supplied to prevent over-humidification. Toilet rooms are exhausted to the outside by gravity through rotary vent heads; the kitchen is exhausted directly to the outside by a propeller fan in the side wall.

Inlets in Ceiling

As inlets in the side wall were practically impossible they were installed in the ceiling with sheet metal diffusers at the openings. The exhaust or return openings, while not ideal, have proved to be satisfactory and while overcrowding in the auditorium and dining room is the

* Consulting engineer, Chicago, Ill.

rule rather than the exception no discomfort has been experienced, as this room is supplied with a five-minute air change.

A total of 19,034 c.f.m. is supplied at an average of 115 F at the inlet when the temperature is 14 F below zero outside. The inlet temperatures for the various rooms vary from 78.6 F in the auditorium to 120 F in such rooms as are supplied only with the necessary air to maintain the desired room temperature.

The designer felt some uncertainty regarding objectionable drafts in the auditorium with a 5-minute air change and an inlet temperature as low as 73 F in mild weather, but as before stated—overcrowding of this room is the rule, and inlet temperatures go as low as 62 F in freezing weather giving almost ideal room conditions.

Water from Deep Wells

Cold city water from deep wells is used in the washer for summer cooling. While no data have been taken on this system for cooling regarding relative humidity, 12 to 18 F difference in temperature may be maintained and no discomfort to the occupants has been noted.

How the System was Designed

Use of Data Sheet

Before we can design the system, we must prepare the data sheet. We must know the heat loss, volume of air required, the inlet temperatures, volume of air returned, and all the various items as called for on the data sheet.

Items 1-2-3 are dimensions and it will be assumed that their solution will be obvious to the reader. *Items* 4-5-6-7 are the constants and factors for the various exposed surfaces. The constants for various walls, ceilings, glass, etc., are given on drawing *A*.

The factor is, in all cases, the constant times the temperature rise. The temperature rise is the difference between outside and the inside temperature, or as the case of the attic space, the difference between the room temperature and an assumed temperature in an unheated space. The Btu loss is the exposed surface in square feet times the factor.

Item 8 is the total in lineal feet of all window and door edge where outside air will leak in.

Item 9 is Btu loss per ft. and is given as 100. This item may vary and if weather-strip is installed, this 100 may be reduced to as low as 10 to 50.

Items 10-11-12 are sq. ft. of exposed surface.

Item 13 is taken from the compass points on the plan, and *item* 14 is room temperature to be maintained.

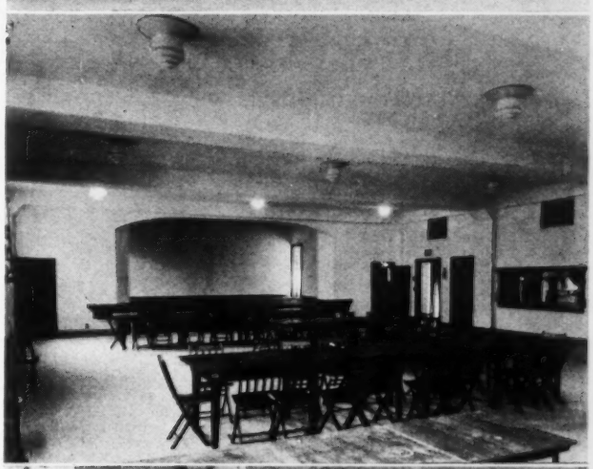
Item 15 is item 12 times factor item 7, and *item* 16 is item 11 times factor item 6. *Item* 17 is total of items 15 and 16.

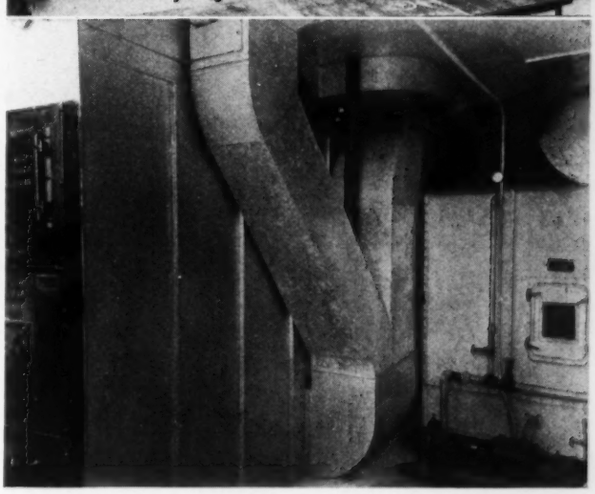
Item 18 is item 17 times the percentages given in the upper left hand corner of the data sheet.

Item 19 will be disregarded as there are no exposed floors.

Item 20 is sq. ft. of ceiling times the factor in item 5. The ceilings on the ground floor are not figured as the rooms above are heated.

Item 21 is item 8 times item 9, and *item* 22 is the total of items 17-18-19-20-21.





Top—The Masonic Temple at Burlington, Wis., in Which a Direct-Fired Hot Blast System Is Installed. Middle—The Main Auditorium and Dining Room Is Rented Out to the Public, for Plays, Dances, Etc. Bottom—Some of the Equipment in the Heater Room. The Air Washer Is Shown to the Right

Item 23 is item 21 divided by two as the wind would blow from only one direction at a time. If the wind is from the south, there would be no leakage in on the north side of the building. *Item* 24 is item 22 minus item 23.

Item 25 is in this instance governed by state law. This building is in the state of Wisconsin, and the ven-

The

N 35 %
NW 30 % NE 30 %
W 25 % E 20 %
SW 15 % SE 10 %
S 5 %

Exposure Factors

Based on -14 deg. Cold Weather & Prevailing Winds From NW.

Bldg. MASONIC TEMPLE Date

Loc.

H. & V.

Archt. Engr.

Rooms No.	1	2	3	4	5	6	7	101	102	103
Use	AUD. & DIN. RM	BILLIARDS	KITCHEN	TOILET	DRESSING	DRESSING	TOILET	LODGE	FOYER	LOUNGE
MEASUREMENTS	62 x 30 10 x 42½ 11½ x 23	34 x 21 9⅓ x 24⅓ 7 x 14						8 x 26	19 x 47	
1. Room Dimensions			37 x 18	15½ x 9½	8 x 8	8 x 8	12'-6" x 8'-6"	46 x 40	4'-6" x 18	22 x 23
2. Cubic Feet Space	29250	13600	8660	1924	832	832	380	36000	11700	6072
3. Room Floor Area	2250	1045	666	148	64	64	106	2000	974	506
4. Floor Const. & Factor										
5. Ceiling Const. & Factor								60 - 27.6	60 - 24	60 - 24
6. Window Const. & Factor	1.2 - 100	1.2 - 100	1.2 - 100	1.2 - 100	1.2 - 100	1.2 - 100	1.2 - 100	1.2 - 123.6	1.2 - 100	1.2 - 100
7. Wall Const. & Factor	.22 - 18.5	.22 - 18.5	.22 - 18.5	.22 - 18.5	.22 - 18.5	.22 - 18.5	.22 - 18.5	.22 - 22.66	.22 - 18.5	.22 - 18.5
8. Exposed Sash and Door Perimeter, equiv.	140	84	28	14	14	14	14		114	28
9. Leakage B. T. U. per ft. of Perimeter	100	100	100	100	100	100	100		100	100
10. Expd. Wall Gross Area	1260	594	198	143	88	176	77	1250	1090	264
11. Expd. Window Gross Area	208	125	50	27	27	27	25	54	356	72
12. Expd. Wall Net Area	1052	469	148	116	61	149	52	1196	714	193
13. Exposure Direction	S.E. & W	N.E.	N	W	W	S.W.	E	S.W.	N.W.	N
14. Room Temperature & mean Temp.	70°	70°	70°	70°	70°	70°	70°	70°	70°	70°
HEAT LOSSES										
15. Wall Loss B. T. U. per Hr.	19462	8676	2738	2146	1129	2756	962	27101	13209	3570
16. Window Loss B. T. U. per Hr.	20800	12500	5000	2700	2700	2700	2500	6674	35600	7200
17. Subtotal Loss B. T. U. per Hr.	40262	21176	7738	4846	3829	5456	3462	33775	48809	10770
18. Expos. Allow. B. T. U. per Hr.	8052	6400	2700	1211	982	820	692	5060	14660	3760
19. Floor Loss B. T. U. per Hr.										
20. Ceiling Loss B. T. U. per Hr.								55200	23376	12144
21. Leakage Loss B. T. U. per Hr.	14000	8400	2800	1400	1400	1400	1400		11400	2800
22. Total Heat Loss	62314	35976	13238	7457	6211	7676	5554	94035	98245	29474
23. Deduct for Heater Size	7000	4200	1400	700	700	700	700		5700	1400
24. Heater Building loss Load	55314	30776	11838	6757	5511	6976	4854	94035	92545	28074
SERVICE										
25. Air Supply C. F. M.	6780	2000	270	153	127	157	114	3600	2000	600
26. Recirculation C. F. M.	6600	1328			113	141		3400	1800	540
27. Air Supply Inlet Temp.	78.6	86.8	120°	120°	120°	120°	120°	94.5	120°	120°
28. Air Supply B. T. U. Service										

ADDITIONAL HEATER LOADS

29. Outside Air Heat Loss, up To room temperature	25902
30. Recirc. Air Heat Loss, up to room temperature	174338
31. Humidity Service	17500
32. Total Item 29	482.997
33. Total Heater Load	199783
34. Chimney Size Required	

tilation code calls for certain definite amounts of air in the rooms that will be used as gathering places by the public. In the dining room the code calls for 3 cu. ft. of air to be supplied for every sq. ft. of floor area. The lodge hall and billiard hall supply are also governed by the code, but the balance of the building may be supplied with the necessary amount of air to properly heat and maintain good air conditions.

Where the volume of air is not fixed by law, we may calculate the cubic feet per minute by dividing the Btu loss (item 22) by the factor 48.96. This will give the amount of air at 120 degrees at the warm air inlet to maintain the desired room temperature, at —14 degrees outside.

Item 26. As we are recirculating most of the air in this installation and item 25 is estimated with air at 120 degrees, and our return air will be approximately 65 degrees, we do not return the same volume in cubic feet.

We determine this volume by the difference in the density of the air at the warm air inlet and the return air opening. In rooms where the inlet temperature is 120 degrees and the temperature at the return is 65 degrees we have: one cubic ft. of air at 120 weighs .068. One cubic ft. of air at 65 degrees weighs .075. .068 divided by .075 equals .90 approximately. C.f.m. at 120 degrees times .90 equals c.f.m. at 65 degrees.

Item 27 is inlet temperature and where we base the c.f.m. by dividing the Btu loss by 48.96 our inlet temperature is 120 degrees. Where we supply more air at a lower temperature as in the dining room we determine the inlet temperature from the formula: Inlet temperature equals [Btu loss divided by (c.f.m. times 1.07)] plus room temperature.

Item 28 is not considered as it is for split steam installations.

Item 29: In estimating our total heat loss, we must consider not only the room loss but also the amount of Btu to raise the air to the room temperature. The loss on this installation will be low as we are recirculating most of the air, and only enough air is taken in from the outside to make up for that exhausted directly from the toilets and kitchen.

Before we can estimate this loss we must know what our volume in cubic feet per minute will be. As the volume in items 25 and 26 are at various temperatures, we reduce the c.f.m. to pounds of air.

Our c.f.m. for item 25 is 19,034 and the average temperature is 115 degrees. One cubic foot of air at 115 weighs .069 pounds. 19,034 times .069 equals 1313 pounds of air.

Item 26 is air at 65 degrees, and as one cubic foot of air at 65 degrees weighs .0756 pounds we have 17,092 times .0756 equals 1292 pounds of air.

1313 minus 1292 equals 21 pounds of air at —14 degrees that must be heated up to room temperature. As one pound of air at —14 degrees is equal to 11.42 cu. ft., we have 11.42 times 21 equals 239.82 c.f.m. As it requires 1.8 Btu to raise one cubic ft. of air from —14 to 70 we have: 239.82 times 60 times 1.8 equals 25,902 Btu per hour for item 29.

Item 30 is 17,092 times 60 times .17 (the Btu required to raise one cubic ft. of air, 65 to 70) equals 174,338 Btu for item 30.

Data Sheet

Note:—It will be noted on data sheet that factors for exposed surfaces are higher for Room 101. This is due to the higher mean temperature in rooms with ceilings over 12 ft. in height.

The following rule is given for the determining of this mean temperature: Subtract 10 from the ceiling height. Multiply the result by 0.017. To this result add the unit of 1 (one). Multiply the answer by the room temperature to be maintained for the mean temperature.

For our problem we have: Ceiling 26 ft. minus 10 equals 16 times .017 equals .272, plus 1 equals 1.272, times 70 equals 89.04 for the mean temperature. —14 to 89.04 equals 103.04 (temperature rise), times constant .22 for wall equals 22.66 for the factor.

104	105	106	107	108	109	110	111	112	113	114	201		
LOUNGE	LADIES	TOILET	TOILET	EXIT	CORRIDOR	FILE RM.	ORGAN	PREP. RM.	COUNCIL	ANTE RM	LOCKER		TOTALS
22x23	9x11	9x8	16x6 9x4	6x7	13'-6"x38	9'-6"x7'-6"	5'-6"x7'-6"	15x9	15x9	14x15 5'-6"x4'-6"	54x12'-6"		
6072	1188	864	1584	504	4617	852	852	1620	1620	2820	8100		
506	99	72	132	42	513	71	71	135	135	235	675		
.60-.24	.60-.24	.60-.24	.60-.24	.60-.24		.60-.24	.60-.24	.60-.24	.60-.24	.60-.24	.60-.24		
1.2-100	1.2-100	1.2-100	1.2-100	1.2-100		1.2-100	1.2-100	1.2-100	1.2-100	1.2-100	1.2-100		
.22-18.5	.22-18.5	.22-18.5	.22-18.5	.22-18.6		.22-18.5	.22-18.5	.22-18.5	.22-18.5	.22-18.5	.30-25.2		
28	14	14	10	20		10	10	14	42	28	40		
100	100	100	100	100		100	100	100	100	100	100		
264	240	96	120	108		114	204	108	288	144	222		
72	36	24	36	21		36	36	36	108	72	45		
193	204	72	84	87		78	178	72	180	72	197		
N	NW	W	W	W		W	SW	S	SE	E	N+W		
70°	70°	70°	70°	70°		70°	70	70°	70°	70°	70°		
3570	3774	1332	1554	1609		1443	3293	1332	3330	1332	4964		
7200	3600	2400	3600	2100		3600	3600	3600	10800	7200	4500		
10770	7374	3732	5154	3709		5043	6893	4932	14130	8532	9464		
3760	2208	933	1288	927		1261	1030	246	1413	1706	2940		
12144	2376	1728	3168	1008		1704	1704	3240	3240	5640	1620		
2800	1400	1400	1000	2000		1000	1000	1400	4200	2800	4000		
29474	13358	7793	10610	7644		9008	10627	9818	22983	18678	18024		
1400	700	700	500	1000		500	500	700	2100	1400	2000		
28074	12658	7093	10110	6644		8508	10127	9118	20883	17278	16024		482997
600	274	160	217	157		185	220	202	480	380	370		19034
540	247			141		166	198	180	423	342	333		17092
120°	120°	120°	120°	120°		120°	120°	120°	120°	120°	120°		115°

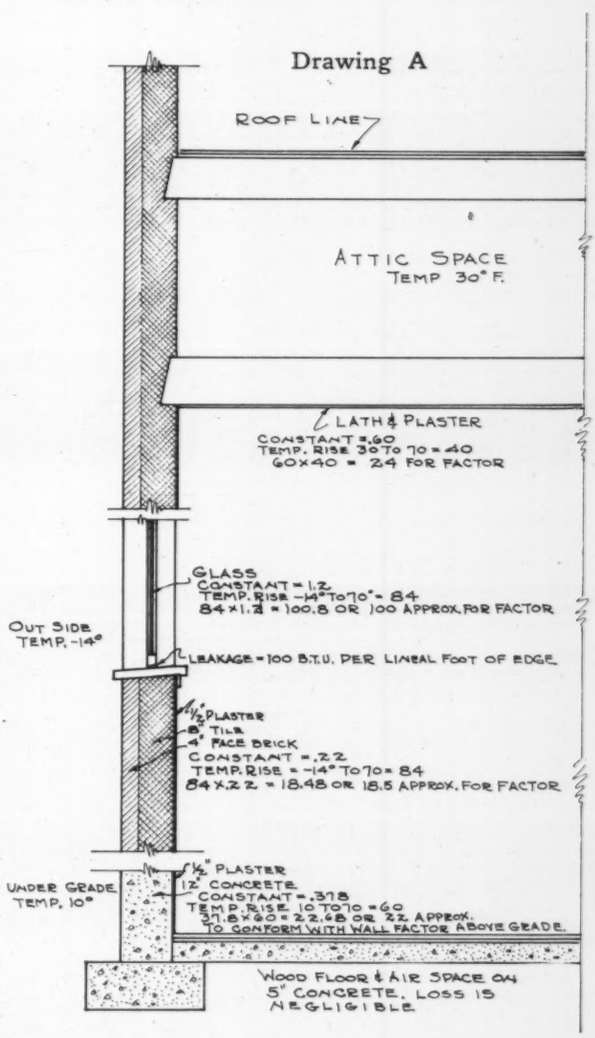
Item 31 we determine from the table:

Temp. Air F	Weight of 1 Cu. Ft.	Cu. Ft. in 1 Lb.	Cu. Ft. 1 Btu Will Raise 1 F	Temp. Air F	Weight of 1 Cu. Ft.	Cu. Ft. in 1 Lb.	Cu. Ft. Btu Will Raise 1 F
0	.0864	11.58	48.77	112	.0694	14.40	60.60
12	.0842	11.87	50.00	122	.0682	14.65	61.60
22	.0824	12.14	51.09	132	.0671	14.90	62.80
32	.0807	12.40	52.20	142	.0660	15.15	63.80
42	.0791	12.64	53.10	152	.0649	15.40	64.90
52	0776	12.88	54.10	162	.0638	15.65	66.00
62	.0761	13.13	55.20	172	.0628	15.90	67.00
70	.0750	13.34	56.30	182	.0618	16.17	68.00
72	.0747	13.39	56.40	192	.0609	16.42	69.10
82	.0733	13.64	57.40	202	.0600	16.67	70.10
92	.0720	13.90	58.60	212	.0591	16.92	71.30
102	.0707	14.14	59.20				

Btu Required to Raise 1 cu. ft. of Air Through Various Temperature Intervals

Outside Temp. F	Temperature of Air in Building						
	40	50	60	70	80	90	100
−30	1.540	1.760	1.980	2.200	2.420	2.640	2.860
−20	1.290	1.505	1.720	1.935	2.150	2.365	2.580
−10	1.051	1.262	1.473	1.684	1.892	2.102	2.311
0	0.822	1.028	1.234	1.439	1.645	1.851	2.056
10	0.604	0.805	1.007	1.208	1.409	1.611	1.812
20	0.393	0.590	0.787	0.984	1.181	1.378	1.575
30	0.192	0.385	0.578	0.770	0.963	1.155	1.345
40	0.000	0.188	0.376	0.564	0.752	0.940	1.128

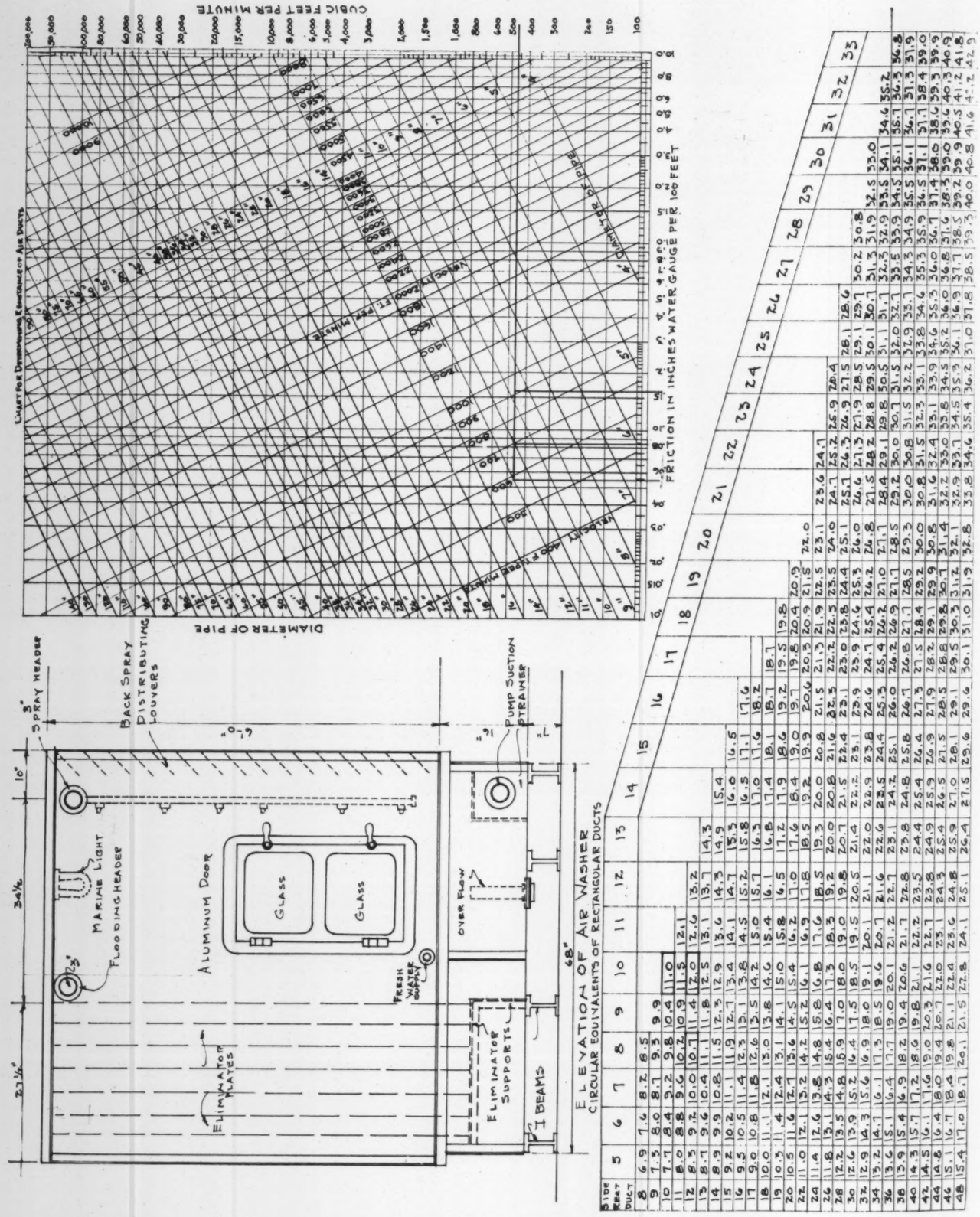
Drawing A


ROOF LINE
ATTIC SPACE
TEMP 30° F.
LATH & PLASTER
CONSTANT = .60
TEMP. RISE 30 TO 70 = 40
60x40 = 24 FOR FACTOR
GLASS
CONSTANT = 1.2
TEMP. RISE -14° TO 70° = 84
84x1.2 = 100.8 OR 100 APPROX. FOR FACTOR
OUT SIDE TEMP. -14°
LEAKAGE = 100 B.T.U. PER LINEAL FOOT OF EDGE
½" PLASTER
8" TILE
4" FACE BRICK
CONSTANT = .22
TEMP. RISE = -14° TO 70 = 84
84x.22 = 18.48 OR 18.5 APPROX. FOR FACTOR
½" PLASTER
12" CONCRETE
CONSTANT = .378
TEMP. RISE 10 TO 70 = 60
37.8x60 = 22.68 OR 22 APPROX.
TO CONFORM WITH WALL FACTOR ABOVE GRADE.
UNDER GRADE TEMP. 10°
WOOD FLOOR & AIR SPACE ON 5" CONCRETE. LOSS IS NEGLIGIBLE

Item 32 is total item 24.

Humidity Control

With the use of air washers, the maintaining of the proper relative humidity is simple. Air at any tempera-



ture passing through a water spray at the proper velocity will become saturated or at a relative humidity of 100 per cent. In this installation we desire 40 per cent relative humidity, and as air at 70 degrees, fully saturated, contains 7.98 grains of moisture per cu. ft., we have 40 per cent times 7.98 or 3.192 grains.

We maintain this amount of moisture by the control of the temperature of the air passing through the washer. As air at 45 degrees will hold 3.414 grains at saturation, we set the thermostat controlling the dampers in the outside air intake at 45 F. This control will be placed at the fan inlet to assure a mixing of the air from the return and the outside. In mild weather when the outside temperature is 45 F or above, cold water

must be used in place of recirculated water or the air will become overhumidified.

Temperature Control

The temperature in the various rooms is controlled by the use of double mixing dampers. These dampers are actuated by the thermostat in the room. As the temperature in the room rises the warm air damper is gradually closed, and as it falls the warm air damper opens. The warm air damper is directly connected to the tempered air damper, and of course works the reverse way.

Diffusion of Air

Warm air inlets are placed in ceilings and sidewalls. The velocity is about 300 ft. per min. at the inlets and the outlets, and where the inlets are in the ceiling, diffusers are installed to keep the air from blowing down on the heads of the occupants.

Determining the Heater Size

We determine the heater size from the formula:

Pounds of coal per hour divided by the combustion rate. Combustion rate equals 13 pounds per hr. per sq. ft. of grate.

We determine the pounds of coal per hr. by the formula:

(Total c.f.m. x 60 x 0.24 x *0.068 x temperature rise fan intake to final temperature) divided by 7200.

Hence for our problem we have: 19,034 $\times$ 60 $\times$ 0.24 $\times$ **.069 $\times$ 70 divided by 7200 equals 184 pounds of coal per hr.

184 divided by 13 equals 14.1 sq. ft. of grate.
7200 equals Btu in one pound of coal.
0.24 equals specific heat of air.
0.069 equals weight of one pound of air at 115 degrees.
70 equals temperature rise, 45 to 115 degrees.
60 equals min. in one hr.

*air at 120 F.

**air at 115 F.

As this installation will have an oil burner the grate surface may vary slightly as the capacity of some heaters may be increased where oil is used as fuel.

Fan and Motor

These may be selected from the catalog of any standard manufacturer and the outlet velocity for this job should be not more than 1700 ft. per min.

Motor to be for available current and to run not more than 1200 r.p.m. at full load.

Determining Resistance Loss

Our static loss will be the sum of the losses in the ducts, washer, and over the heater. We estimate the pressure loss for the duct system on the longest duct. In this plan it will be the duct to *room 113*.

We measure the duct for length and for the various sizes. We have 108 ft. of 12 x 10. To this we add 10 ft. for each 90-deg. elbow. We have 2-90 deg. and 4-45 deg., or a total of 4-90 deg., or 148 ft. of 12 x 10 to carry 480 c.f.m.

Our riser is 16 ft. of 10 x 10. Our basement run is 14 ft. of 8 x 12 with 2 long elbows equivalent to one 90-deg., and one short 90-deg., or a total of 34 ft.

We now refer to our chart for determining the resistance in air ducts. As the chart reads in round ducts, we refer to the table of circular equivalents of rectangular ducts for equal friction per lineal ft. We find our 12 x 10 duct is equal to a round duct 12 in. in diameter.

We now locate 480 c.f.m. on the chart at the right hand side and follow through to the left until we intersect with the 12-in. diameter line and here we drop to the base of the chart and find .054 for our resistance in 100 ft. of pipe. As we have 148 ft. we have 148 times .054 or .0799.

Our riser is 10 x 10 and equals a duct 11 inches round. For this duct we have a resistance of .082 for 100 ft. As we have but 16 ft., we have $\frac{16}{100}$ of .082 or .0131.

Our 8 x 12 duct in basement is equal to a round duct of 10.7 in., and our resistance is .152 for 100 ft., or .0516 for 34 ft.

Our total is .0799 plus .0131 plus .0516 equals .1446 for the supply ducts. Our resistance for the returns we place as the same as our supply or .15 approximately. Our resistance over the heater will be .25 inches (taken from heater table), and through the washer .3 (from manufacturer's data). Hence we have a total of .1446 plus .15 plus .25 plus .3 equals .84 of an inch for the system. It must be understood that this is just an estimate and the actual loss may vary. The final test may show .75 to .85.

Design of Duct System

As we must maintain individual temperature control in each room, we must run a separate duct to each room as the temperature in the various ducts will vary.

For this installation the ducts were sized to velocities. Where the duct leaves the plenum chamber, the ducts were sized for a velocity of 700 to 800 ft. per min., in the risers 600 to 700 ft. per min., and in the attic ducts 600 to 800 ft. per min. These are the velocities for the small ducts. The larger ducts may be designed for a higher velocity, 800 to 1000 ft. at the plenum, 800 to 900 in the risers, 600 to 800 in the attic ducts. The larger the duct the greater the allowable velocity.

Lower velocities must be maintained in the returns, and they should not exceed 500 ft. per min., except in the return chamber.

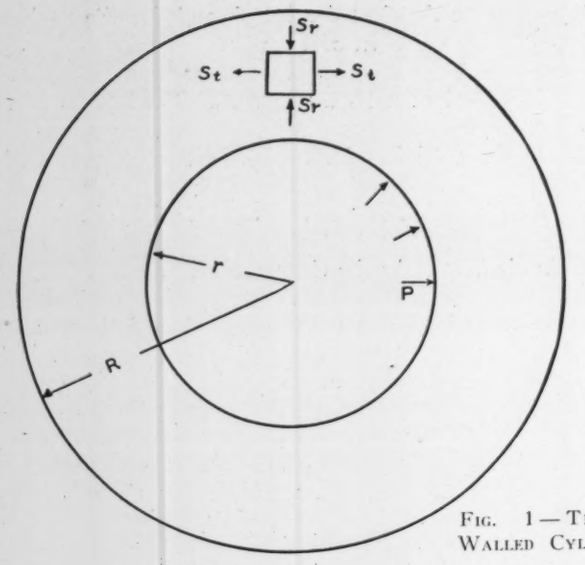


FIG. 1 — THICK-WALLED CYLINDER

By Harvey A. Wagner*

When theoretically correct results are desired, it is the writer's belief that Clavarino's formula should be used for designing piping, where closed end conditions prevail. This article compares the Birnie and Clavarino formulas and shows their differences in application. It is a helpful contribution to piping engineers, as it discusses the fundamentals of design clearly.

Bursting Stress in

High Pressure Piping

Fundamental Formulas of Design

A RECENT article [1] regarding the effect of higher pressures and temperatures on piping touched upon the design of pipes and cylinders as affected by higher pressures. The trend toward higher pressures both in central station and industrial power plants requires the piping designer to become familiar with the fundamental principles involved in the rational selection of a formula suitable for the design of thick walled pipes or cylinders.

Comparison of Birnie and Clavarino Formulas

Mr. Harman selected Birnie's formulas as most suitable to meet the conditions encountered in modern piping work. While the writer will grant that Birnie's formula results in thicker pipe walls than Clavarino's formula when applied to the same problem, the use of Barlow's formula results in still thicker walls, yet the latter cannot be considered as theoretically correct for piping work. Where closed end conditions are prevalent as in piping work, Clavarino's formula is theoretically correct and in the writer's opinion should be used where exact results are desired. If for any reason it is desired to allow a greater factor of safety, this should be done by designing for a lower working stress in Clavarino's formula, which seems preferable to using an inexact formula which errs on the side of safety. The use of Barlow's formula often is justified, however, from a standpoint of ease in solution. The writer intends to show, by the following discussion, the fundamental theoretical differences between the Birnie and Clavarino formulas as well as their difference in application.

* Engineer, The Detroit Edison Company, Detroit, Mich.

[1] "Higher Pressures and Temperatures as They Affect the Design, Installation and Maintenance of Piping" by J. J. Harman, December, 1930, HEATING, PIPING AND AIR CONDITIONING, p. 985.

Application of Birnie Formula

The correct application of the Birnie formula may be illustrated by the condition met in hydraulics of a pipe connecting two reservoirs with no intervening valve. Neglecting the action of friction of the water when flowing, a true condition of an open ended pipe is obtained. This is somewhat analogous to the conditions existing in gun barrels for which application Birnie's formula was developed.

Any particle of metal in the wall of a thick cylinder subjected to internal pressure is influenced by at least two stresses, a radial stress produced directly by the internal pressure and a tangential, or hoop stress. By reference to Fig. 1, it can be seen that the tangential tension acting simultaneously with the radial compression both tend to produce a lengthening in a tangential direction and a shortening in a radial direction. These two stresses result from internal pressure, assuming the pipe or cylinder to have open ends, or ends supported independently of the pipe walls.

The intensities of these stresses may be determined by the following formulas which are due to Lamé and which are based on the assumptions that the material is isotropic and homogeneous.

Referring to Fig. 1, in which

p = unit internal pressure, lb. per sq. in.
r = internal radius, inches.
R = external radius, inches.
R_x = radius at which stress is desired, inches.
S_t = tangential stress produced, lb. per sq. in.
S_r = radial compressive stress produced, lb. per sq. in.

$$S_t = \frac{pr^2}{R^2 - r^2}\left(\frac{R^2}{R_x^2} + 1\right) \quad (1)$$

$$S_r = \frac{pr^2}{R^2 - r^2}\left(\frac{R^2}{R_x^2} - 1\right) \quad (2)$$

If the ends of the pipe are attached solidly to the pipe walls, a stress is also produced in an axial direction which tends to shorten the radial dimension and also the tangential dimension of the particle. Any actual pipe line approaches this state, which should be considered the limiting condition in design. Lamé's equations as given above consider each stress as acting independently. The limit of elastic resistance, however, must be reached through a combination of these stresses. Theories regarding their combination do not agree so that it is necessary to rely upon test results to indicate the correct theory.

Theories As to When Elastic Failure Is Produced

Briefly, the three principal theories as to when elastic failure is produced are:

(1) For a certain value of the maximum principal stress (Rankine).
(2) For a certain value of the maximum principal strain (St. Venant).
(3) For a certain value of the maximum shearing stress which is proportional to the greatest difference between principal stresses (Guest).

Rankine and St. Venant Theories

The Rankine theory does not consider the strengthening or weakening effect of stresses at right angles to the principal stress. If this theory were true, the fracture of materials would occur along a plane in a direction at right angles to the maximum stress. Experiment has shown that for ductile materials, this is not the case. In addition, it is known that homogeneous materials withstand hydrostatic pressure above the breaking value as indicated by the Rankine theory. For ductile materials, therefore, the Rankine theory is not considered favorably. The St. Venant theory assumes that the elastic strength of a material in which the maximum stress is tensile, for instance, is influenced by other stresses in mutually perpendicular planes which also may be present.

Guest Theory

The Guest theory assumes that elastic failure occurs in the nature of sliding or shearing produced by the combination of the maximum and minimum stresses obtaining. An indication of the distribution of the three stresses over the wall of a cylinder with closed ends under internal pressure is shown in Fig. 2.[2]

Results of Tests on Thick-walled Cylinders

Considerable test work on thick-walled cylinders has been carried out by the Army Ordnance Department at Watertown Arsenal[3] as well as elsewhere.[4] Its re-

[2] From *Strength of Materials*, by A. Morley.
[3] *Combined Stresses in Thick-Walled Cylinders*, E. B. Norris. Trans. *A. S. M. E.* A. P. M.-51-6, 1929.
[4] *Destruction Test of 66-in. Forged Steel Penstock Pipe*, J. L. Cox. Trans. *A. S. M. E.* IS-50-3, 1928.

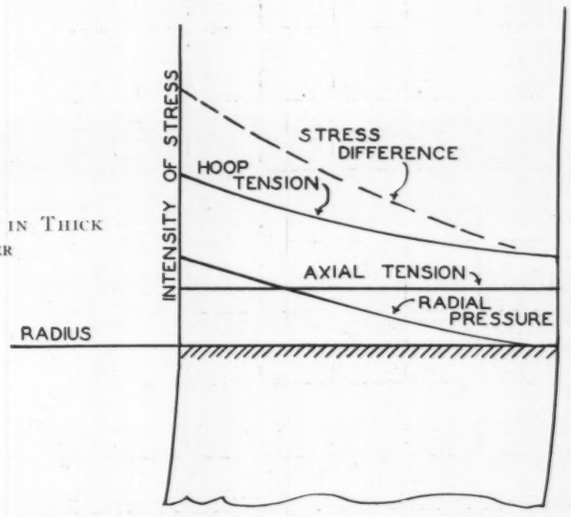
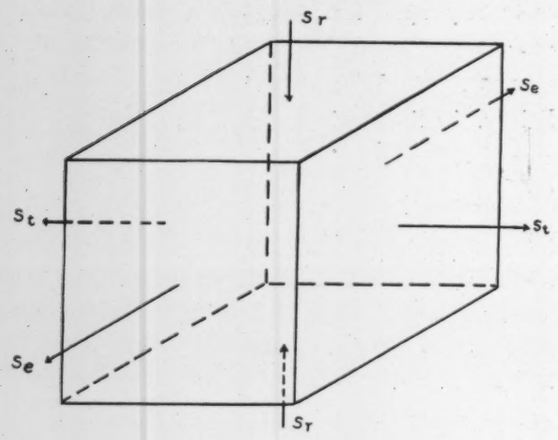


FIG. 2—STRESSES IN THICK CYLINDER

FIG. 3—STRESSES ON PARTICLE



sults with open end cylinders have led it to conclude that for cases of tension and compression in mutually perpendicular planes such as encountered by a particle in a thick-walled cylinder, the theory of St. Venant is correct. In cases where the internal pressure was increased to such a point as to cause rupture of the cylinder, the fracture showed no evidence of failure by shear. This constitutes an experimental check on the validity of the St. Venant theory.

Closed Cylinder with Ends Attached to Walls

Assume the particle of Fig. 3 to be one in a thick-walled cylinder. Assuming a closed cylinder of some length with ends attached to the pipe walls at a considerable distance from the point of stress analysis, the three stresses produced by internal pressure are represented by S_t, S_r and S_e. S_t represents the tangential or hoop stress, S_r the direct radial stress produced by internal pressure and S_e, the longitudinal stress produced by the internal pressure acting against the cylinder ends. According to the theory of St. Venant, if θ represents the elastic limit of the material in tension, the limit of elastic resistance is reached when

$$\frac{S_t}{E} + \frac{S_r}{mE} - \frac{S_e}{mE} = \frac{\theta}{E} \qquad (3)$$

or

$$S_t + \frac{1}{m}(S_r - S_e) = \theta \qquad (4)\ \text{(Clavarino's condition)}$$

where $1/m$ represents Poisson's ratio and E represents the modulus of elasticity. If it is assumed that the ends are independently supported, equation (3) becomes

$$\frac{S_t}{E} + \frac{S_r}{mE} = \frac{\theta}{E} \qquad (5)$$

or

$$S_t + \frac{S_r}{m} = \theta \qquad (6)\ \text{(Birnie's condition)}$$

It readily can be seen that the left hand member of equation (6) is greater than that of equation (4). These two equations represent the fundamental theoretical difference between the formulas of Clavarino and Birnie respectively. The combination of equation (6) with equations (1) and (2) results in Birnie's formula, which may be stated as follows in terms of external and internal diameters, D and d respectively, assuming a value of 0.3 for Poisson's ratio,

$$S = p\left(\frac{1.3\,D^2 + 0.7d^2}{D^2 - d^2}\right) \qquad (7)\ \text{(Birnie)}$$

Clavarino's equation, combining (4), (1), (2) and $S_e = \dfrac{pd^2}{D^2 - d^2}$ may be stated,

$$S = p\left(\frac{1.3\,D^2 + 0.4d^2}{D^2 - d^2}\right) \qquad (8)\ \text{(Clavarino)}$$

It may be observed as pointed out above that the stress as computed by Birnie's formula will be greater than that computed by Clavarino's formula, other things being equal.

Clavarino Formula Theoretically Correct

In piping work, the condition most often encountered is that of a pipe with closed ends, which may result

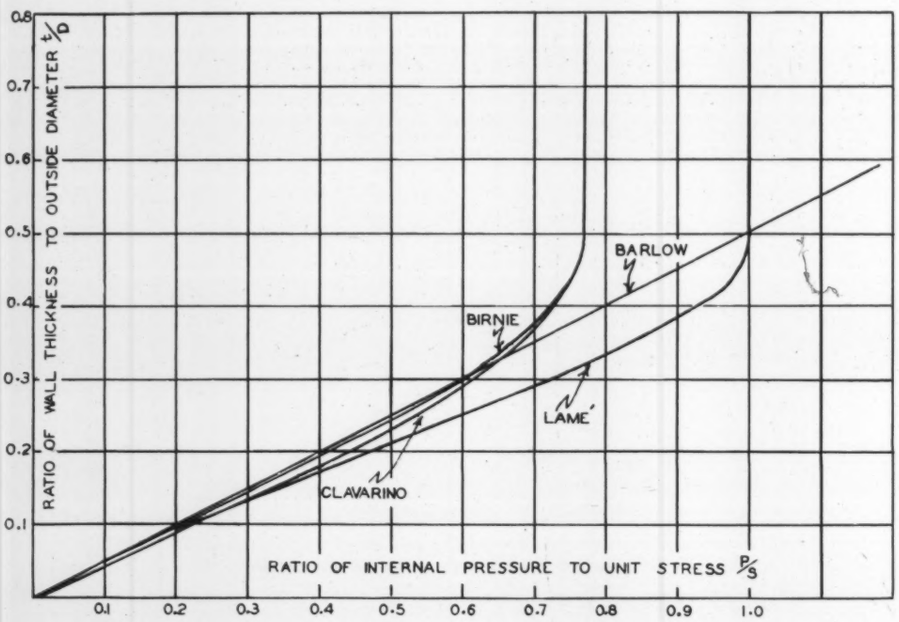


FIG. 4—COMPARISON OF INTERNAL FLUID PRESSURE FORMULAS

in a pipe line from closing a valve. For this condition, the formula of Clavarino is theoretically correct and should be used. Tests[5] have shown that for conditions where Clavarino's formula applies, test results agree very closely with computed values. As shown in Fig. 4, for t/D ratios up to approximately 0.15, any one of the formulas mentioned may be used satisfactorily.

Practical Limit

It is of interest to note from Fig. 4 that it is impossible to design a simple cylinder to withstand a fluid pressure beyond a certain value for a given material. This may also be seen from equation 8, which may be transposed as follows:

$$P/S = \frac{D^2 - d^2}{1.3\,D^2 + 0.4d^2} \qquad (9)$$

To obtain the maximum allowable value of P/S, assume $d = 0$ or the condition of an infinitely thick cylinder (solid). For this condition, the value of $P/S = 1/1.3$ or 0.769. The practical limit is reached at a lower value of P/S inasmuch as the constant 0.769 represents the condition of a solid cylinder and is of no practical importance with respect to piping problems.

[5] *National Pipe Standards*, 1924.

Use Barlow Formula Up to t/D = 0.3

The writer agrees with Mr. Harman's conclusions that for ordinary purposes and for ranges of t/D up to 0.3, the Barlow formula gives results well within the scope of error justified by the use of commercial pipe and tubing where variations in strength of material, wall thickness, temperature stresses, etc., are too great to warrant the use of theoretically correct formulas such as those of Clavarino or Birnie. Where theoretically correct results are desired, however, in the writer's opinion Clavarino's formula should be used for piping work where closed end conditions prevail.

Author of Original Article Replies to Above Discussion

By John J. Harman*

MR. WAGNER'S article clearly shows the fundamental differences between the Birnie and Clavarino formulas and is also very helpful in calling attention to the fact that recent laboratory research on thick walled cylinders, conducted by the Army Ordnance Dept. at Watertown Arsenal, confirms St. Venant's maximum-strain theory of the combination of tensile and compressive stresses in mutually perpendicular planes where the tensile stress is the greater of the two. The stresses created in thick walled cylinders by internal pressure are of this character and the tests very definitely show the correctness of the St. Venant theory in this case. In view of the fact that it has been customary to use the St. Venant theory for combining stresses in problems of this character, it is very reassuring to have this supporting experimental evidence.

Fig. 3 of Mr. Wagner's article illustrates the stresses created by internal pressure in a closed end cylinder and it will be noted that the axial tension caused by the internal pressure acting on the closed ends tends to produce a shortening of the particle of metal in the tangential direction. This action opposes the lengthening of the particle in the tangential direction caused by the combined action of the tangential and radial stresses, reduces the resultant tangential stress and increases the load carrying capacity of the cylinder. By the same reasoning it is equally clear that if the cylinder is subjected to a longitudinal compression from some outside source equal to or greater than the longitudinal tension created by the internal pressure, the load carrying capacity of the cylinder would not be increased over that of an open end cylinder and might actually be reduced.

How Often Do Closed End Conditions Prevail?

The writer fully agrees with Mr. Wagner's conclusion that for exact results when closed end conditions prevail Clavarino's formula should be used for computing the stresses, but feels that there is considerable question as to how frequently in pipe line work we really have this condition. When a valve is closed at the free end of a pipe line, this condition undoubtedly prevails and, neglecting the effect of temperature changes, the illustration used by Mr. Wagner of "a pipe connecting two reservoirs with no intervening valve" represents an open end cylinder condition in which the stresses would be correctly computed by the Birnie formula. However, the great bulk of piping lies in between these two extreme cases. When a valve in a continuous pipe line is closed, it appears to the writer that the end thrust of the internal pressure would in most cases be carried by the pipe anchors or by the equipment to which the pipe is connected. It might be argued that in this case the portion of the pipe on the pressure side of the valve would act as a closed end cylinder and beyond the valve there would be no internal pressure, but the first assumption is not true, because the free longitudinal movement of the pipe is actually restrained. In case this valve were open instead of closed, the end thrust of the internal pressure would surely be carried at some other point in the system.

* Assistant Chief Engineer, Walworth Company, Boston, Mass.

Expansion Sets Up Compressive Stresses

Another important factor that prevents the frequent occurrence of true closed end cylinder conditions in piping work is the effect of temperature changes. Many pipe lines carry fluids that have been heated above atmospheric temperature and this tends to lengthen the pipe due to thermal expansion. If, as is usually the case, the pipe is anchored at intervals, the expansion sets up compressive stresses that neutralize the benefit of the axial tension produced by the internal pressure acting against a closed valve.

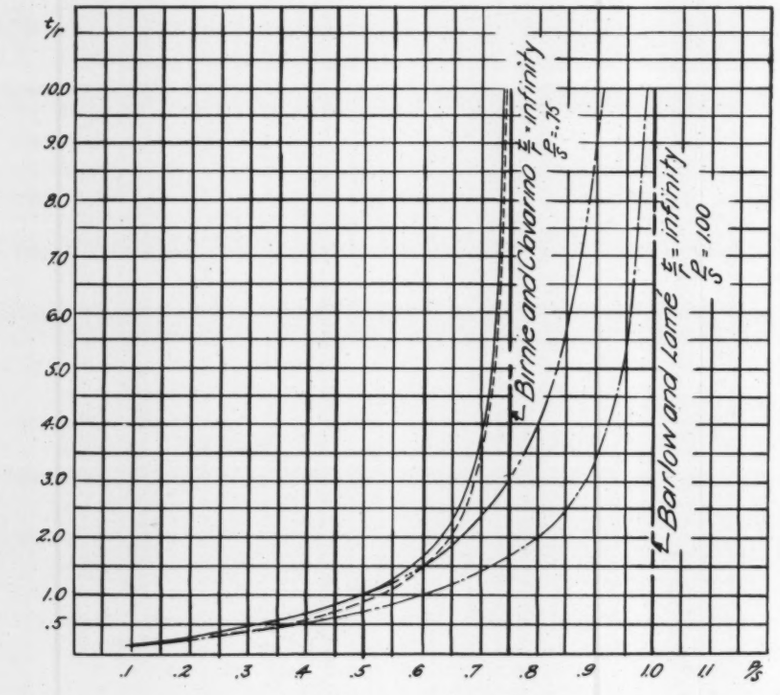
Piping Work Analogous to Open End Conditions

It consequently would appear that the usual stress conditions in piping work are more nearly analogous to the open end than to the closed end cylinder conditions. Due to the fact that no structure is stronger than its weakest link, the writer feels justified in defending his original statements, namely: "————————it is more conservative for pipe line computations to use Birnie's open end cylinder formula which gives higher stress values than the closed end cylinder formula of Clavarino . . . . . for $\frac{P}{S}$ ratios up to ½ the simple Barlow formula gives values practically in agreement with the more accurate Birnie formula, although erring slightly on the side of safety. . . . . The more complex Birnie formula, therefore, finds its economical field of application in the zone where the $\frac{P}{S}$ values exceed ½, and the simple Barlow formula can be used without hesitation for $\frac{P}{S}$ values up to ½, which covers practically all the usual piping problems encountered in industry." Suitable allowances to cover manufacturing limitations, etc., should, of course, be made in applying any of these formulas to an actual design.

Explains Difference in Charts

It may be well in passing to explain some of the obvious differences between the original article and Mr. Wagner's article, the reasons for which may not be readily apparent. The slight difference between Mr. Wagner's formulas (7) and (8) and the Birnie and Clavarino formulas as given by the writer are due to the fact that slightly different values for Poisson's ratio were used in the two cases.

The marked difference in the appearance of the curves shown in Fig. 5 of the original article and Mr. Wagner's Fig. 4 is due to the fact that the ordinates of the original set of curves represent the ratios of wall thickness t to



COMPARISON OF BARLOW, BIRNIE, CLAVARINO AND LAMÉ FORMULAS FOR t/r RATIO UP TO 10

inside radius r, while in the latter case the ordinates of the curves represent the ratios of wall thickness t to outside diameter D. Values of the $\frac{t}{r}$ ratio range from zero to infinity, which makes a chart with wide vertical spacing, while the $\frac{t}{D}$ ratio has a low limit value of zero and a high limit value of 0.5 which represents the condition when the cylinder becomes a solid bar and $t = \frac{D}{2}$

In making the original chart the writer felt that the curves showing the relation between $\frac{P}{S}$ and $\frac{t}{r}$ more clearly illustrate the effect of variations in cylinder dimensions, internal pressure and allowable stress in the metal. When the wall thickness of a cylinder is progressively increased so as to resist an increasing internal pressure, at the same time keeping the capacity (inside diameter) of the cylinder constant, both r and S are constants and the $\frac{t}{r} : \frac{P}{S}$ ratio curve really represents the true effect of the internal pressure P on the required wall thickness t.

The drawing in this discussion is introduced to show the relation of the Lamé, Clavarino, Birnie and Barlow formulas when platted in this way.

Mr. Wagner's article very justly points out that when theoretically correct results are desired, simple formulas not strictly exact are unsuitable for use even though the error involved may be very slight, but he apparently agrees with the writer that for the ordinary computations encountered in present day piping systems the Barlow formula gives results so nearly exact that the errors may be neglected. As compared with the Birnie formula the errors are practically nil in the range up to $\frac{P}{S} = \frac{1}{2}$ and $\frac{t}{r} = 1$.

Air Conditioning

for Egg Storage

A Complex Problem

By C. E. Baker*

PROPER air conditioning of egg storage rooms is one of great importance. It presents several problems that are found in air conditioning for the storage of few other commodities.

The degree of humidity is exceedingly important and the range over which satisfactory storage conditions prevail is narrow. If the relative humidity of the air in egg storage rooms becomes too high, molds and mildews develop, causing the eggs to become musty and to have an unpleasant taste. If, on the other hand, the relative humidity of the air becomes too low, moisture is taken from the eggs, causing them to shrink and "rattle."

The Temperature and Humidity Relation

The temperature relation also is important. Eggs freeze at about 28 F and are likely to burst their shells from the expansion of the frozen contents. A satisfactory egg temperature, therefore, is 1 or 2 degrees above their freezing point, provided the temperature may be accurately controlled. At this temperature it is difficult to maintain a sufficiently high relative humidity in many houses and shrinkage is more likely to be a problem. At low temperatures, of course, the development of mold and mildew is reduced to a minimum. As the temperature rises it is easier to maintain a satisfactory humidity but the development of molds is encouraged. At temperatures of 29 to 31 F a relative humidity of 85 per cent is considered safe, while at a temperature as high as 40 F the humidity must be held at 60 per cent or below.

Susceptible to Odors

Eggs are susceptible to storage odors and will absorb any odor carried in the air. Frequently they absorb the odor of the wooden case or the strawboard pockets or fillers to such an extent that quality is greatly reduced. During the storage period a natural, gradual decomposition of the egg takes place and as a result gaseous impurities are given off. If these are permitted to accumulate they eventually become objectionable and injurious to the produce.

What Air Conditioning Does

Air conditioning consists of air changing, to remove injurious gases, air circulation, to provide a means of bringing fresh, sweet air in contact with individual eggs and humidity control, to maintain the required relative humidity.

Fresh air must be introduced into the storage rooms in sufficient amount to remove odors and to keep injurious gases at a low concentration. The fresh air must be in rapid motion in order to penetrate the rather tight cases and change the air about each individual egg. For this reason air conditioning systems that employ circulating fresh air and by which accurate humidity control is possible are claimed to be very satisfactory for egg storage rooms. Where no such system exists a frequent change of air should prove beneficial, but forced circulation within the room generally is considered a necessity.

The stacking of the cases in such a way as to facilitate thorough circulation of air is another important consideration. Ventilated cases and perforated fillers also aid in efficient circulation.

Chemical Factors Influencing Keeping Qualities

The chemical factors influencing the keeping quality of hen's eggs has been studied in detail by Sharp[1] and other investigators. Three important changes in the structure take place under unsatisfactory storage conditions: (1) The white loses its thick, jelly-like consistency and becomes watery. This condition is known as "watery whites." (2) The passage of water from white to yolk causes a more fluid condition of the yolk. (3) The membrane surrounding the yolk weakens, permitting the yolk to flatten or break when the egg is broken.

Several investigators have found that as soon as the egg is laid the *pH* of the white begins to increase, following the loss of carbon dioxide. This change has been found to be from *pH* 7.6 in the case of a fresh egg to *pH* 9.7 at the end of an average storage period. This represents an increase in alkalinity of about 100 times. The yolk is slightly acid in a fresh egg, having a *pH* of 6.0. This also increases in *pH* value but not so rapidly nor to so great an extent as does the white.

Effect of Carbon Dioxide

It is commonly recognized that the decomposition of proteins is accelerated by high *pH* values and this, also,

* Agricultural Experiment Station, Purdue University, Lafayette, Ind.

[1] *Ice & Refrigeration*, March, 1930.

has been found to hold true with eggs. When eggs are stored in an atmosphere containing small amounts of carbon dioxide above the normal content, carbon dioxide is taken up by the egg to maintain an equilibrium between the egg and the surrounding air. Some of this carbon dioxide taken up by the egg combines with water to form carbonic acid which neutralizes an equal amount of alkalinity.

The concentration of carbon dioxide in the storage air necessary to maintain this equilibrium and hold the white at approximately the same *pH* value as in the fresh egg, depends upon the temperature of the storage. As the temperature decreases, the required percentage concentration lessens proportionately. At 30-31 F, about 3 per cent of carbon dioxide in the atmosphere is reported to be required to maintain this balance and prevent the increase in alkalinity of the egg.

The importance of the carbon dioxide relation is coming to be recognized to the extent that the introduction of the gas into the storage rooms to build up the desired concentrations is being recommended by some investigators. Even when used far in excess of the required amounts, there is no disagreeable effect upon workmen who handle the produce. Compressed gas from cylinders is one convenient source, as it may be metered out in accurate amounts. Sharp estimates that if sufficient CO_2 were added to a storage room once each week for six months to give an atmospheric concentration of 1 per cent, the cost would be less than one cent per case of eggs.

As carbon dioxide is slightly heavier than air, a moderate continuous circulation would serve to maintain a uniform distribution of carbon dioxide in the storage atmosphere.

Air Purity of Economic Importance

By F. E. Hartman†

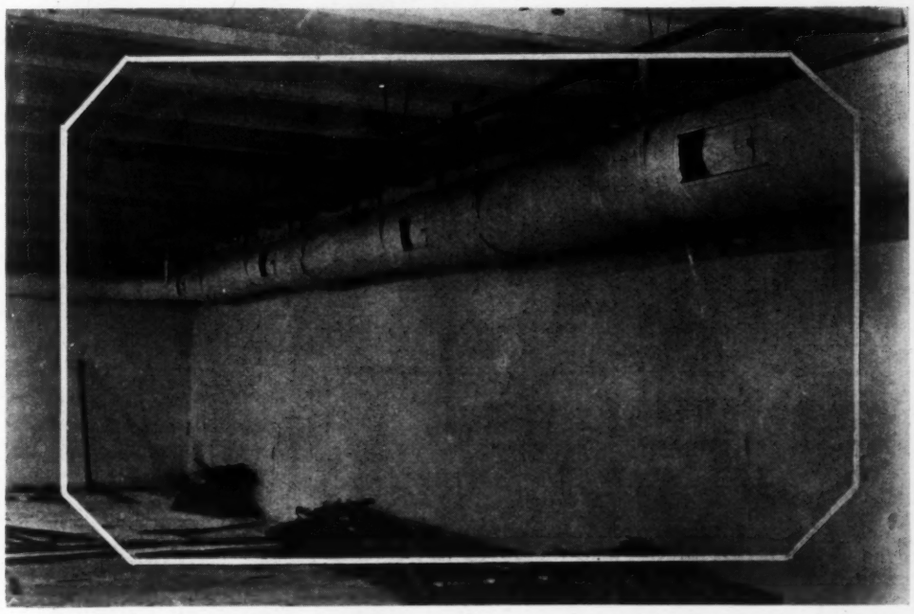
PERHAPS nowhere is the purity of air of greater economic importance than in the storage of eggs. Eggs readily absorb the gases given off by building materials, cratings, and fillers. In addition, the eggs themselves give off partially decomposed gases, which, if allowed to accumulate, will become reabsorbed, and together with the other foreign gases of the air greatly affect the flavor of the eggs.

Fortunately the gases affecting eggs in storage are in a state of low oxidation and are readily susceptible to attack by ozone. Ozone completes the oxidation of these gases, oxidizing them to such innocuous products as carbon dioxide and water vapor. With the use of ozone, in definite and controllable quantities, the air in a storage room can be maintained free from gases affecting eggs.

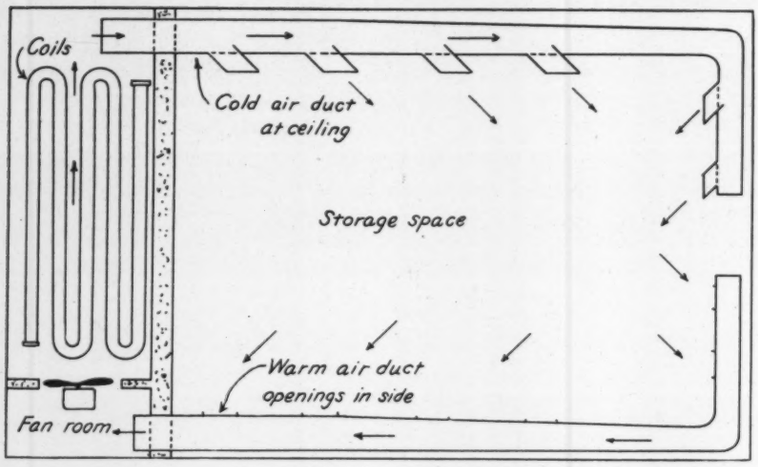
† Director of Research, Biological Engineering Laboratory, Chicago, Ill.

Proper Circulation Needed

A factor in air conditioning for egg storage is humidity. Eggs should be stored so as to insure good ventilation of all crates, preferably with positive means for



An Egg Storage Room Showing the Warm Air Return Duct. No Coils or Bunkers Are Used in the Room. The Pipe in Front of the Air Duct Is a Fire Sprinkler Line. The Lath Have Been Used Between Cases of Eggs in Stacking to Form Air Spaces to Aid in Thorough Circulation. Indiana Refrigerating Co., Indianapolis



A Common Method Used in Circulating Air in an Egg Storage Room. The Air Is Blown Through a Bunker of Refrigerated Coils at One End of the Room From Which It Enters the Cold Air Duct and Is Forced Out Through Openings Along the Duct. The Warm Air Is Pulled Into a Similar Duct Along the Opposite Side of the Room Near the Ceiling, Carried Across the Coils and Recirculated

maintaining air motion in and about the crates. Water readily evaporates from eggs, due to diffusion of vapor through the shells; as the condition of the vapor within the shell is saturated, the rate at which evaporation takes place will be governed by the degree of saturation of the air surrounding the eggs.

Evaporation produces shrink and rattle, which immediately brings the quality of the egg into question. In order to prevent shrinkage, a fairly high humidity should be carried and when the degree of saturation that will prevent shrink is approached, the danger from mold and mildew becomes greatly increased. Here ozone finds an economic application. A concentration of 0.4 part per million of ozone in air will inhibit the growth of all molds and mildew commonly encountered in packing and storage houses.

Ozone does not affect eggs until a concentration manyfold that required for air purification and mold inhibition is reached. The apparatus ordinarily used for producing ozone for use in cold storage can not, even when operated continuously at full capacity, produce a condition that will in any wise become harmful to the eggs. This safety factor is inherent in the chemical instability of ozone, as an ozone concent[illegible]ot build up due to the continuous decomposition of the ozone supplied the rooms.

Cost of Using [illegible]

From the point of view of cost, 100,000 cu. ft. of

Eggs in Storage Should Be Stacked so as to Permit the Circulation of Air Between and Around the Cases. Indiana Refrigerating Co., Indianapolis

cold storage space can be maintained at the proper concentration of ozone, both for air purification, and inhibition of molds, at a gross expenditure of 150 watts per hour; no special attendant is required. Ozone is finding an ever widening application in cold storage and many plants in the United States and Canada are now equipped with controlled ozone apparatus. The haphazard use of ozone is not recommended, as the results will not be constant.

A Brief Discussion

By John H. Ruckman††

I understand that it has been found that eggs contain a considerable quantity of carbon dioxide and that when exposed to air even at the proper humidity, this gas tends to leave the eggs, causing them to shrink and probably slightly damaging the flavor. For this reason I understand that storage houses are now adding carbon dioxide to the air in the storage rooms. The Seymour Packing Co., which has gone about as far as any in following up this work, keeps most accurate records of both temperature and humidity conditions in the storage rooms. I understand that a degree of precision has been attained which permits the classification of egg according to the size of the air bubble and that quantities have been sold to specification on this basis.

Mr. Hartman's paper brings in some matters which are less well understood at least by the majority of those who will read his paper. The writer has been unable to locate any plant in which ozone has been tried as an air purifier. There is no question, however, that ozone would be effective in killing fungus growth or molds. Unless it has been actually demonstrated that the rather penetrating odor of the ozone is not transmitted to the egg, it would be well to consider this phase of the matter.

The question might also arise as to whether or not this very powerful chemical reagent would affect the egg tissues. I presume that all these matters have been given attention by Mr. Hartman and if tests have been made on any of these points, I would be very glad to hear of them and the results obtained.

†† Consulting engineer, Topeka, Kansas.

Cleaning Routine for Air Filters at Compressor Intake

A 10-unit battery of air filter cells is installed at the intake of a 5,000-cubic foot heavy-duty air compressor at the Akron plant of the Firestone Tire & Rubber Co.

The compressor was installed to supply about one-third of the compressed air requirements. It is a two-stage unit rated to handle 5,000 cubic feet of free air per minute. Air is delivered from the 36½ x 27 inch low-pressure cylinder at 30 lb. gage pressure and from the second stage (22 x 27 inches) at 100 lb. The driving motor is rated at 822 h.p., and the normal demand of 4,500 c.f.m. imposes a load equal to about 95 per cent of the nominal motor capacity.

As originally installed, this unit had no air filters of any kind. The intake duct was carried to the roof of the power house and left entirely open there at a point near the vent from a pulverizing mill which handles about 1000 tons of coal daily. The effect of the dust-laden air was immediately evidenced. Compressor capacity dropped rapidly during the first six weeks of operation and an inspection showed that its valves and ports were so heavily incrusted with grease, soot and dirt as to make a thorough cleaning imperative.

Millwrights regularly employed about the plant handled the erection and installation of the filter equipment. This consisted of setting up a rectangular air chamber, closed at the top and bottom and equipped with suitable supporting racks for the filter cells on two sides and one end. The original down-duct, leading the air to the compressor, was connected into the other end of the air intake chamber.

Cleaning consists of removing two dirt-laden cells daily and replacing them immediately with two of four spare cells, which have previously been cleaned in a soda-ash solution and freshly dipped in a specially-mixed oil. With ten cells in use, the two-cell daily replacement means that the entire battery is gone over every five days. This is more often than usual, but is accounted for in this case by the unusually dirty condition of the air going through and by the fact that the compressor operates almost continuously for 24 hours a day.

It is necessary to clean the compressor twice a year. This is a two-day task for a 10-man crew and is usually done on Sundays when the air requirements can be met temporarily by smaller compressor equipment. The eight high-pressure valves and three of the large low-pressure valves are usually dismounted, cleaned and replaced on the first Sunday. The nine remaining valves on the low-pressure side are similarly treated on the next Sunday.

In the manufacture of linseed oil, linseed oil is used as the hydraulic medium. Mr. Fletcher describes the process in this article and includes information for the piping engineer and contractor

Linseed Oil Manufacturing Process Depends on

An Extensive Piping System

By J. Gordon Fletcher*

LINSEED oil is made from flaxseed, or linseed, which is grown in North America, South America, India and Russia. Most of the crushers on the Atlantic seaboard depend on the Argentine crop for their seed supply.

Comparatively few of the users of linseed oil know how it is manufactured. Piping engineers are using it constantly as the vehicle for white lead, red lead and many of the common pipe-joint compounds.

As the manufacturing process depends upon an extensive piping system in which linseed oil itself is used as the hydraulic circulating medium, it is worthy of a brief study by those interested in piping.

After the seed has been unloaded from cars or ship it is carefully screened in order to remove foreign seeds, dirt, stones or particles of iron. It is then passed through a set of chilled iron rolls, generally five rolls to a set, with each one weighing about a ton, which reduce the seed to a fine, dry meal.

The meal is then passed to the top of a steam jacketed cooker which consists of three or four large pans set one above the other. An agitator keeps the meal in constant motion as it gradually passes down from the upper to the lower sections through small bottom gates. It requires approximately one and a half hours for the meal to travel down through the cooker, meanwhile its temperature is being raised to 200 F by steam at a pressure of 45 pounds in the jackets.

* Asst. Supt., John T. Lewis & Bros. Co., Philadelphia, Pa.

Steam economy is secured by covering the cooker sections with blocks of 85 per cent magnesia insulation protected by a canvas jacket.

In tempering one bushel of seed in the cooker 9 lb. of steam are required or approximately 2 lb. of steam per gallon of oil pressed. These are average figures taken over a considerable period of time, as results secured from short runs mean little because changes in the nature of the seed necessitate frequent adjustments of the steam flow.

Underneath the cooker is located a former, or moulding table, with a counter-balanced head. When this head has been raised a hair cloth is laid upon the table and then a small buggy, actuated by a hydraulic plunger, spreads a charge of meal over the cloth. The head is then pulled down on the table and locked after which a large hydraulic ram lifts the table against the head and compresses the meal into a compact slab. After the ram has been lowered again and the former head lifted the ends of the cloth are folded over and the slab is removed by means of a light steel pan underneath it. The slab then is inserted between the plates of a vertical press which has a capacity of 22 cakes.

The press is equipped with steel plates about 36 in. by 18 in. which are lifted by a ram 16 in. in diameter. The ram travel is controlled by a change valve with carefully timed chokers. Upon opening the valve oil at 500 lb. pressure is admitted to the cylinder and the ram rises rapidly. As soon as oil begins to flow from the meal the 500-lb. line closes automatically and the pressure gradually builds up to 4200 lb. When the oil flow ceases the cake is removed and the edges, which have a high oil content, are trimmed in a machine equipped with two hydraulic rams operating at 500 lb. pressure.

Linseed Oil as Hydraulic Medium

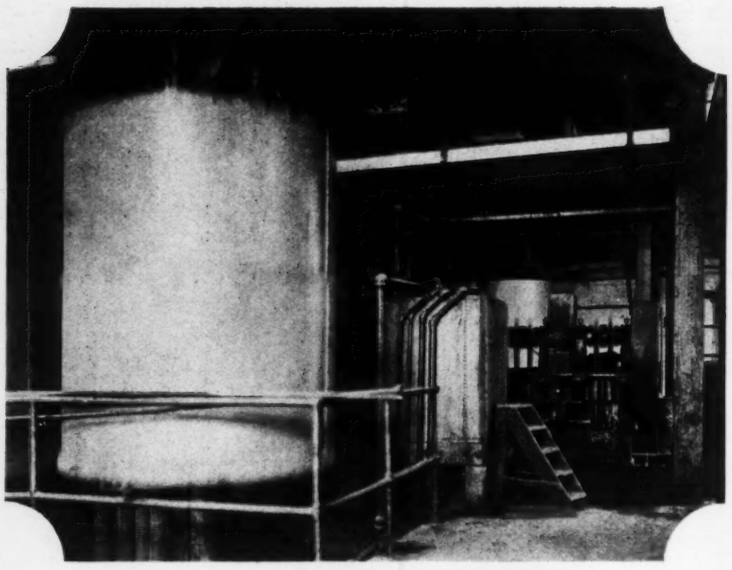
The hydraulic system consists of two complete pressure lines with returns to a common storage tank, linseed oil being used as the hydraulic medium. This eliminates the possibility of the oil being contaminated with water, a condition to be guarded against at all times. The 4200 lb. pressure is required for presses only while the cake former, presses and trimmer utilize 500 lb.

Oil is fed by gravity from the storage tank to the pressure pumps, each one of which has two high and two low pressure cylinders. Uniformity of pressure is maintained by means of 20-ton accumulators floated on each line and so weighted as to give the required pressure. A safety valve incorporated in each accumulator control system furnishes protection against excess pressure. Sudden surges in the lines are minimized by shock absorbers which consist of chambers in the line fitted with pistons working against heavy coil springs, thus acting to cushion out any sudden fluctuations.

The Pumps

Pumps are of the single-acting plunger type, vertical or horizontal, and driven through belts or direct connected motors. Speed reduction gears are necessary as the pumps make only 60 rpm, the high pressure end delivering 3 gallons per minute and the low, 15 gallons. Two pumps will supply 24 presses without any trouble. It can readily be seen that the problem is not one of handling large quantities of oil or of working with large

Cooker, Former, Press, Change Valve and Part of the Piping Are Shown in This View



THIS VIEW SHOWS TWO PUMPS (SUCTION SIDE), THE PUMP STORAGE TANK, BOTH ACCUMULATORS WITH SOME OF THE PRESSURE LINES AND THE SHOCK ABSORBERS. THE SHOCK ABSORBERS ARE THE NESTS OF COIL SPRINGS UNDERNEATH THE HIGH PRESSURE ACCUMULATOR, WHICH IS AT THE TOP OF ITS STROKE

sizes of pipe as the high pressure pump outlet is 1 in. and the low 1½ in.

In none of the press-room operations is a quick transfer of a large amount of oil called for. The fastest motions are found in the cake former but here the displacements are small. It requires a period of 18 minutes to bring the press ram up to 4200 lb. pressure and during this time 33 gallons of oil are pumped into the press cylinder.

Presses Operated in Groups

Presses are operated in groups of six and the charging of them is so timed that but one of the presses in a group is receiving the full flow of oil at a time.

The press change valves are so adjusted that six minutes elapse after the 500-lb. pressure line is opened before the 4200-lb. line goes into service. In order to get this adjustment of the choker valves, frequent checks of the timing are made with a stop-watch. In the six minutes during which the 500-lb. pressure operates the ram stroke is almost completed and therefore the oil demand is a maximum. Very little oil is required in completing the stroke under 4200 lb. pressure.

It takes a pressman at least eight minutes to strip and charge one press, whereas the press he has just charged completes its maximum oil demand in six minutes. As the pressman moves along the line there can be, therefore, but one press in each group at a time drawing any considerable amount of oil.

In this way fewer pumps are required and smaller pipe sizes are permissible. The accumulators act to supply any excessive demand and as long as packings are kept tight no trouble is experienced due to lack of capacity. It is advisable to provide a spare pump for emergency service.

Through the use of linseed oil as the hydraulic medium possibility of corrosion is eliminated, but it is necessary to provide an adequate method for filtering the circulating oil in order to prevent cutting of the valves by dirt and chips. This filtering is best accomplished by a screen placed in the storage tank in front of the pump suction outlet. This is an important detail because some of the change valve choker openings are measured in thousandths of an inch and their enlargement due to abrasion upsets the press timing.

The Piping

Knowing the pipe size outlets to be furnished by the manufacturer on the presses, cookers, pumps, etc., the total number of units to be installed and the operating cycle, it is possible to proceed with the pipe layout. Two pumps will take care of twenty-four presses and four cookers. The pump high pressure outlet is 1 in., low pressure 1½ in.; the press high pressure connection is ¾ in. With press operations staggered and the maximum demand for oil reduced, a 1½ in. line is sufficient for the high and 2 in. for the low pressure lines.

For 4200-lb. pressure double extra heavy pipe is required and it is inadvisable to use unnecessarily large sizes as some difficulty is met with pipe of this weight in larger diameters; there is a tendency for it to split along the butt welds.

Forged steel fittings and valves are used for the high pressure lines. Bolted flanged couplings with ground joints are favored for linseed oil lines although the forged steel pipe unions now on the market where tried out in this service are giving good results. This union is a stock article with most supply houses.

The use of acetylene or electrically welded joints for this service is not favored as yet principally because any pin hole leaks at such high pressure are most difficult to eliminate after the lines are filled with oil. The use of a welding torch on piping containing a film of linseed oil and coated outside with oxidized oil and meal dust, as is usually the case, presents a fire hazard calling for extra precautions.

As welding technique improves and skilled welding operators become more plentiful there is no question but that there will be applications of heavy-walled seamless tubing with all-welded construction in linseed mills.

A linseed mill operating 24 hours a day needs constant maintenance. Properly installed, with good pipe and fit-

tings, high pressure lines should last for years without any replacements.

Lines Sectionalized

For the 500-lb. system extra-heavy pipe and extra-heavy iron fittings will answer the purpose although forged steel fittings are often justifiable. Heavy brass valves or cocks are favored for sectionalizing these lines and are necessary because the cake formers and trimmer require frequent shut-downs for repacking.

The number of shut-off or sectionalizing valves placed in each system depends upon preference and plant policy, also upon the size of the press room. Each press change valve is equipped with chokers which are used as shut-off valves to permit packing or regrinding. If it is desired to operate but a part of the presses then a sectionalizing valve on each group is preferable.

As oil in the pressure lines circulates at a temperature but slightly above that of the room little or no trouble is caused by expansion or contraction. All piping must be rigidly fastened to prevent any movement due to line surges. This is particularly true of the discharge lines from the accumulators to the storage tank, as quite a heavy surge is set up when the shut-off valve trips.

In a hydraulic system, particularly one where the pressure is furnished by plunger type pumps, when the valve operations occur suddenly, there is a continual tendency for hammering to develop in the pipe lines. The force produced is sufficient to split pipes and fittings which normally are strong enough to hold the operating pressures.

"As the linseed oil manufacturing process depends upon an extensive piping system in which linseed oil itself is used as the hydraulic circulating medium, it is worthy of brief study by those interested in piping."

Particularly heavy shocks are produced in the accumulator lines, and these must be carefully supported. In a recent installation pipes between the accumulators and the pump supply tank broke repeatedly. The trouble was not eliminated until all of the lines had been re-run and clamped rigidly so that any movement was impossible.

Shock relief valves should be installed in the pipe line next to the accumulator and at the end of the pressure line.

In another instance a cake-former pressure line snapped off because it had been improperly run and the constant hammering finally cracked the pipe.

If the building design and press room arrangement are not too complicated a neat pipe layout is possible because of the small sizes used. Of course the line should be as straight as possible with a minimum of fittings; pipe bends are often used. Every effort should be made to reduce the possibilities for leaks as each one means a shut-down of the mill, therefore good materials and workmanship are absolutely necessary. It must be kept in mind by those in charge of operation and maintenance that a repair requiring but a short time will probably throw the whole mill system off schedule for a much longer period as all operations are on a piece work basis calling for uninterrupted mechanical service.

"It must be kept in mind by those in charge of operation and maintenance that a repair requiring but a short time will probably throw the whole mill system off schedule for a much longer period, as all operations are on a piece work basis calling for uninterrupted mechanical service."

Clean Lines Essential

Clean pipe lines are essential to efficient operation. Serious trouble is sure to follow if dirt, iron particles or meal are allowed to accumulate in them. Before filling with oil blowing the pipes with air will remove some

of the dirt which will collect during construction. Nothing but filtered oil should be put into the lines. After the mill has been running for a while the maintenance engineer will find it necessary to clean out the pump supply tank, change the filter screen in it, and regrind all of the choker and accumulator valves.

The screen in the supply tank should be of the twin type to permit cleaning without shutting down. The proper maintenance of such a screen is difficult because of the fine foots which continually close the mesh but the beneficial results gained justify the effort.

Failure to keep a sufficient supply of oil in the pump tank will cause the entire hydraulic system to fill with air, the elimination of which will be most difficult and time-consuming. It is surprising how small a quantity of air will shut down the pressure system.

Make-up Oil

The pump supply tank should be large enough to hold considerable more oil than would be required to fill the system, in order to provide for leakage at the pumps and presses. At stated times filtered oil, in measured quantities, is run into this tank to make up for the losses. The amount added in this way must be accurately checked in order to determine the true oil yield.

It is usually not desirable to admit this make-up oil by means of an automatic float valve because the tank level fluctuates so constantly that the valve tends to hold open unnecessarily.

It is possible to use a low-level alarm if preferred. As all of the oil lines should be located above ground, an excessive leak would be noticed in time to prevent a dangerous drainage of the system.

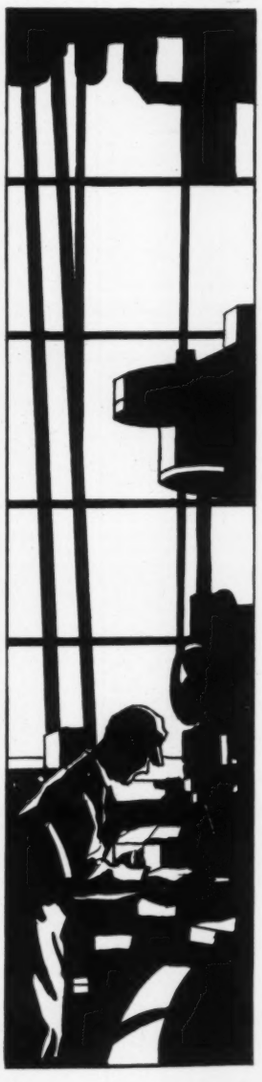
Linseed oil is entirely satisfactory for use as a hydraulic medium, as it keeps the lines perfectly clean. Pipes recently removed after forty years of service were found to be in excellent condition. By using filtered oil all of the meal is eliminated and there is little or no chance for foreign materials to get into the system if the pump supply tank is kept well covered.

Water in the oil would not interfere with the hydraulic operation, but would make the oil useless as a paint vehicle. There is always a small percentage of moisture in fresh oil, as the meal contains water, but this is removed before the oil is shipped to the consumer. Linseed oil becomes very expensive as a hydraulic medium if there happen to be any leaky underground pipes draining into a sewer or the earth. Where overhead joints develop small leaks the oil eventually oxidizes and forms gummy streamers, which do not add to the appearance of the lay-out.

Packing

The press, former and pump cylinders are packed with leather crimps or combination packings which are very easily damaged by foreign particles in the oil. A plunger once scored is most difficult to keep from leaking at such high pressures.

The use of leather crimps as packings for linseed oil machinery plungers is general, as they are easily made on the job by using the proper molds. Also, some of the asbestos and flax packings will give satisfactory service. On the pressure pump plungers leather crimps are standard, but some asbestos compositions hold well. All-metal packings may prove satisfactory if installed with new plungers, but there is some objection to their use on the ground that metal chips can get into the system and score the valves. This point is debatable.



For successful operation of a linseed oil mill the pipe lines must be of good materials, carefully installed and kept full of clean oil. Attention to this assures satisfactory service.

Metallurgical Requirements for High Temperature Steam Piping†

By F. W. Martin*

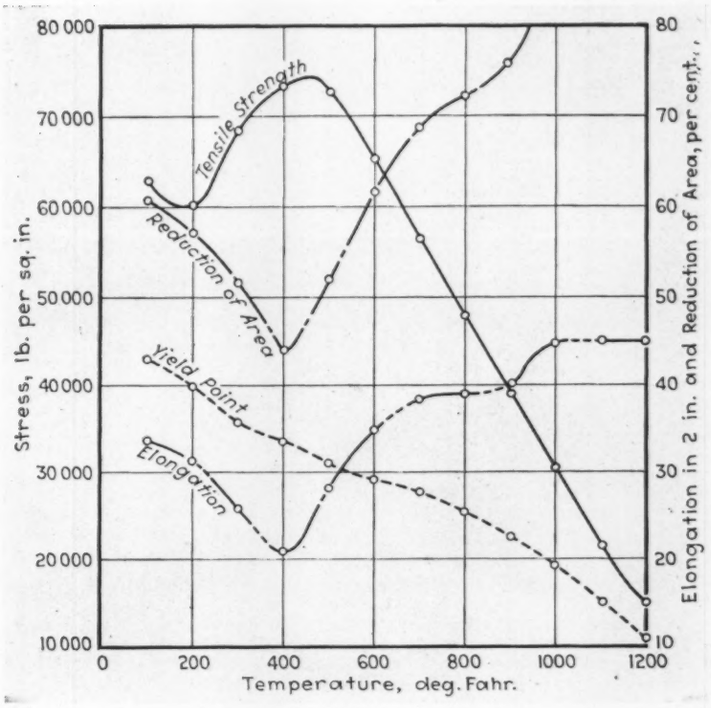


FIG. 1 — SHOWING TENSILE PROPERTIES OF FORGING STEEL

The section of the recent A. S. M. E. —A. S. T. M. high temperature symposium devoted to power plant requirements was divided into three parts: boilers, piping and turbines. Mr. Martin's paper deals with the piping; included under this heading are pipe, valves, flanges, fittings, bolts or studs, nuts, and the component parts of valves—bodies and bonnets, seat and disk rings, and stems.

SOME years ago the maximum steam pressure in a power plant was 200 lb. with a temperature of 480 to 500 F. Under these conditions low-carbon steel pipe with cast-iron valves and fittings was fairly satisfactory. Ordinary bolts were used to hold the joints together. Leakage was prevented by soft gaskets. Questions of expansion, creep, embrittlement, etc., were practically unknown. Just before the World War, however, the steam pressure began to increase. The Joliet Power Station was built in 1917 for a steam pressure of 300 lb. and a temperature of 600 F. These high pressures and temperatures were continued after the war. Steam power plants are in operation at this time with steam pressures up to 1,200 lb. and temperatures up to 750 F. There are plants under construction to operate at temperatures as high as 850 F, with a pressure of 1,200 to 1,400 lb. A small experimental plant to operate at a

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† From *Symposium on Effect of Temperature on the Properties of Metals* presented at joint meeting of *A. S. M. E.* and *A. S. T. M.*, June, 1931, at Chicago.

temperature of 1,000 F and a pressure of 400 lb. is nearly completed.

In Europe, pressures as high as 3,200 lb., with a temperature of 700 F, have been used. In this country temperatures as high as 1,400 or 1,500 F and pressures as high as 2,500 to 3,000 lb. have been considered. At these high temperatures the physical characteristics of metals are quite different from those at ordinary temperatures. The tensile strength and elastic limit are much lower and tend to approach each other. The elongation for a given load is much larger and the hardness is decreased, so that either design stresses must be considerably lower than at ordinary temperatures, or the materials used must possess much higher physical properties.

From the standpoint of standardization, simplicity of design and reduction of manufacturing cost, the preferable solution is the use of stronger material and the retention of present design stresses for the higher temperatures. The present steel flange standards used by practically all piping and fitting manufacturers were designed for use up to and including 750 F when made up in carbon cast steel with a minimum yield point of 36,000 lb. per sq. in. The range of operating temperatures above 750 F can very practicably be divided into about three groups: 750 to 850 F, 850 to 1,000 F, and temperatures over 1,000 F. With this as the basis of procedure there will be one set of standard dimensions with one set of patterns and dies for all pressures and four grades of material depending upon the temperature.

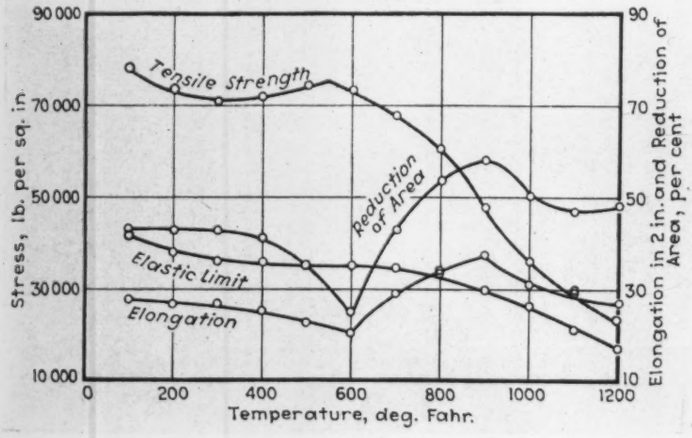


FIG. 2 — SHOWING TENSILE PROPERTIES OF CARBON CAST STEEL OF THE FOLLOWING COMPOSITION: CARBON, 0.30 TO 0.35; SILICON, 0.34 TO 0.39; MANGANESE, 0.74 TO 0.80; SULPHUR, 0.026 TO 0.033; PHOSPHORUS, 0.018 TO 0.026 PER CENT

Pipe

The first item to be considered is pipe. At the present time most designers have limited themselves to seamless steel tubing with a minimum tensile strength at room temperature of 62,000 lb. per sq. in. and other physical properties similar to those shown in Fig. 1, for all pressures above 400 lb. per sq. in. and for temperatures up to and including 750 F. The design of this material has been based on the modified Barlow formula as given in the *A. S. M. E.* boiler code, using a unit stress of 9,000 lb. per sq. in. Some designers have possibly gone to the higher stress value of 12,000 lb. per sq. in., recently allowed in the boiler code.

The limiting factors in pipe design are those which will cause failure by bursting or leaking. It should be remembered that this paper deals only with steam power plant piping, where the temperature throughout is approximately uniform and there are no hot spots, thin spots or heavy scale to cause concentration of stress at any point. Such conditions are liable to occur in boiler tubes, superheater tubes, oil refinery stills and other places where the tubes or pipes are subjected to external heating or contain a corrosive material. Under such conditions entirely different design limits must be employed.

A great deal has been said in the last few years about creep in metals at high temperatures. This only applies to pipe in so far as it may cause a reduction in the thickness of the pipe wall to the point of failure. A small growth in the diameter of the pipe is of no consequence, since it comprises no moving parts and has nothing with which to interfere, and any increase in length which might occur due to creep is so small that it is of no consequence. Power plants are designed for a useful life of at least twenty years. Therefore, the allowable increase in size of the pipe is such that during the life of the equipment it will not reach the bursting point.

Short-time tension tests of pipe, tubing, cast-steel fittings, etc., show a total elongation at the time of failure of 25 per cent or more. Approximately 10 per cent of this takes place before serious necking down of the section starts. If the total deformation is kept below this amount, there is little danger of failure. Assuming that failure by creep will occur at approximately the same total elongation as failure in the ordinary short-time tension test at the temperature under which the material is used, a total increase in pipe or fitting circumference of 5 to 6 per cent could be allowed with safety. A total creep of 5 per cent in 200,000 hours would give an increase of 0.5 in. in the diameter of a 10-in. pipe, which would not be practical. The actual permissible creep, therefore, would be less than the minimum necessary from a safety standpoint. A working range of 2 to 3 per cent would probably be permissible.

Fittings

In pipe installations there are many places where several connections must be made close together, or where a change in direction is necessary and pipe bends are not convenient. Present practice utilizes fittings of cast steel, with approximately the physical properties shown in Fig. 2, for such service where the temperature is not

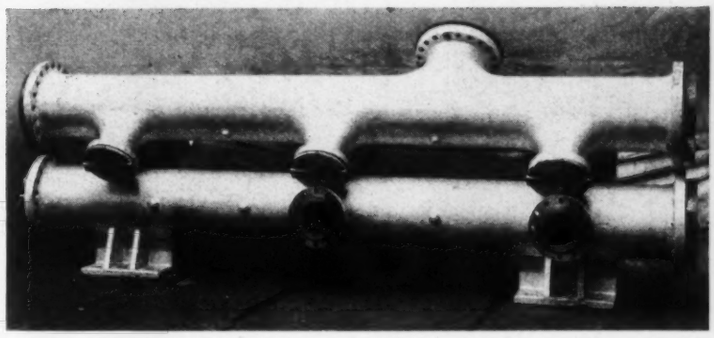


FIG. 3—HEADER FOR SERVICE WHERE THE TEMPERATURE IS NOT OVER 750 F

over 750 F. Fig. 3 shows a header for such service. This material is in the annealed or normalized state and under no condition is given a liquid quench. This corresponds to *A.S.T.M.* standard specifications for carbon-steel castings for valves, flanges and fittings for high-temperature service (A 95-29). The minimum dimensions are those given in the steel flange standard and are based on a working stress of 7000 lb. per sq. in. with 50 per cent increase in thickness to compensate for irregular shapes. Forged steel with a minimum yield point of 36,000 lb. per sq. in., corresponding to *A.S.T.M.* standard specifications (A 105-28) has also been used extensively, especially in small sizes for service up to 750 F.

It is essential that the materials to be used at higher temperatures for this service be reliable and lend themselves to safe design. The tensile strength, elastic limit, etc., at the temperature at which the material is to be used should be at least as high as the corresponding quantity for cast carbon steel at 750 F. The material must cast well; that is, it must form sound castings in comparatively thin-walled sections and must be free from blow holes, shrinkage cracks, foreign inclusions or porosity. The creep rate at the temperature in question, 850 and 1000 F, is subject to the same considerations as pipe. In addition, the material must be readily machineable at room temperature, as smooth contact faces must be provided. The loose flanges on the pipe constitute a large field for both cast and forged material.

For service at 850 F at least one firm is using a chromium-nickel alloy cast steel with the physical properties shown in Fig. 4; the curves of short-time strength indicate that this material is stronger at 850 F than carbon-steel castings are at 750 F, and just as strong at 1000 F as carbon steel at 750 F.

The small steam plant mentioned before to operate at 1000 F is being built with the well known 18 per cent chromium, 8 per cent nickel alloy for the castings as well as the pipe. This installation will be watched with a great deal of interest.

Valves

The valves for high temperature and high pressure steam lines present several problems. A generally used type is the gate valve. This valve consists of a body, bonnet and disk of cast material, two seat rings in the body and two disk rings of rolled material, a stem, a packing gland and packing around the stem as well as a stem bushing. The body and bonnet are fastened together by studs and nuts with a gasket or ring in the joint.

The body is made of the same material as fittings for the same service. The creep and other stress conditions and limitations, however, are more severe than in the case of fittings. The valve contains movable parts which must make a tight fit when closed.

When heated all parts of the valve expand. The expansion of the seat rings screwed into the body must be very nearly equal to the expansion of the body, otherwise

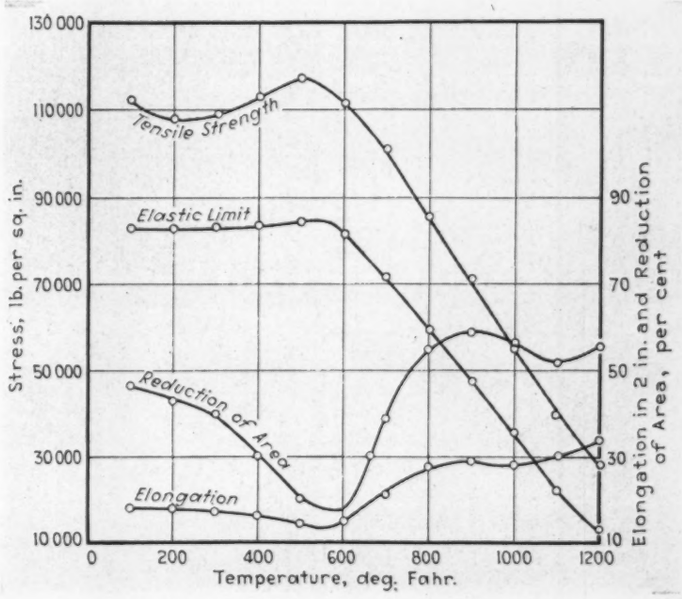


FIG. 4—SHOWING TENSILE PROPERTIES OF ALLOY CAST STEEL. NICKEL, 1.22; CHROMIUM, 0.63; CARBON, 0.40; SILICON, 0.02; SULPHUR, 0.03; AND PHOSPHORUS, 0.03 PER CENT

they will become loose and steam will leak behind them, or they will force themselves out of shape and bulge, thus preventing the tight closing of the valve. The rings on the disk present the same problem: if they expand less than the disk material, the rings will become loose and may fall out; if they expand too much they will bulge and prevent tight closing.

The seat rings must also possess a high resistance to galling and scoring. This may or may not be a direct function of hardness. Experience has indicated that certain metal combinations show a high resistance to scoring and galling at suitable operating temperatures. If the seat metals seize and gall badly the valve will not be tight and may also be hard to open.

When the valve is closed the disk is subjected to the pressure of the steam over its entire area. The seat rings in both the body and disk must be sufficiently strong in compression at the operating temperature to withstand this load. The metals in the seat rings should be capable of withstanding a compressive load, at the operating temperature, of at least 9000 lb. per sq. in. to meet this condition. The metal must be practicable to machine in the annealed condition in order that screw threads may be cut and surfaces finished. After machining, the metal should harden without appreciable distortion when subjected to suitable heat treatment.

The stem must possess good resistance to torque, must be easily machined in order that good screw threads can be cut in it, and must take a polished finish so it will pass back and forth through the packing without seizing and without leaking. It must possess compressive strength at the operating temperature sufficient to withstand a load of at least 9000 lb. per sq. in., since it must close the valve under load. Creep in the stem is not of great importance, since a small elongation of the stem would merely lower the disk an equal amount. The stem must be corrosion resistant, otherwise the threads will rust together.

Another factor in valve stem life is the fact that with the valve open the stem never is anywhere near the temperature of the fluid in the pipe if that is at a high temperature.

Studs and Nuts

The various sections of pipe, as well as the valve body and bonnet, are held together by studs with a nut on each end. This type of fastening is used in preference to bolts because of the stress concentration under the bolt head and the difficulty of making suitable alloy-steel bolts.

The use of carbon-steel bolts in high temperature steam pipe lines has been strongly discouraged because of the tendency of carbon-steel bolts to become brittle when used for a long period at high temperatures. As a result, the ordinary carbon steel has been replaced by alloy steel for high temperature installations. Up to 750 F, material conforming to *A.S.T.M.* standard specifications for alloy steel bolting material for high temperature service (A 96-97) has proved satisfactory. One alloy very widely used is *S.A.E.* 3135 or 3140 suitably heat treated to give the physical properties shown in Fig. 5, which conform to the above specifications.

The dimensional standards for steel flanges and flanged

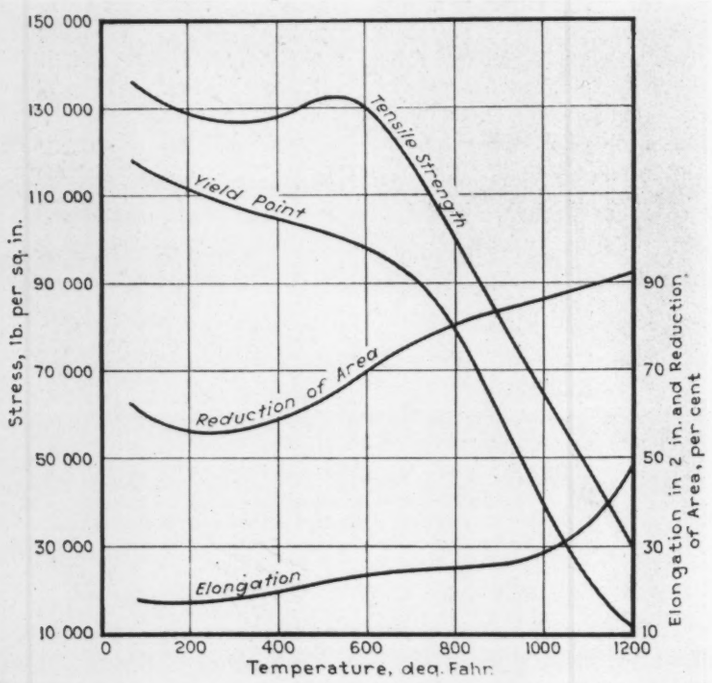


Fig. 5—Showing Physical Properties of *S.A.E.* Bolting Steel No. 3140 at Various Temperatures. The Results Are Averages of Seven Series of Tests. Chemical Composition: Carbon, 0.35 to 0.45; Manganese, 0.50 to 0.80; Phosphorus, 0.04 Max.; Sulphur, 0.05 Max.; Nickel, 1.00 to 1.50; Chromium, 0.45 to 0.75 Per Cent

fittings in *A.S.A.* are designed with a stress in the studs, due to steam pressure only, of not over 9000 lb. per sq. in. As a matter of fact very few of the standards actually show a stress due to steam pressure over 7000 lb. per sq. in. It should be remembered, however, that this is only a small part of the total load on the studs. It has been found by experiment that with gasket joints a total gasket pressure ten to twelve times the steam pressure is necessary to hold the joint tight. Also, experiments have shown that steam fitters, in tightening up joints, will put from 30,000 lb. per sq. in. to 65,000 lb. per sq. in. stress into the studs. Usually from 30,000 to 35,000 lb. is a fair figure. Material for studs for high temperature service must possess high strength and good elongation, so that the studs will not snap off when highly stressed during the heating up of the steam line. At this time the studs are subjected to a very high stress because the flanges and pipe heat up much more rapidly than the studs, due to the low rate of heat conductivity between the studs and the flanges. Just how much this excess load amounts to is uncertain.

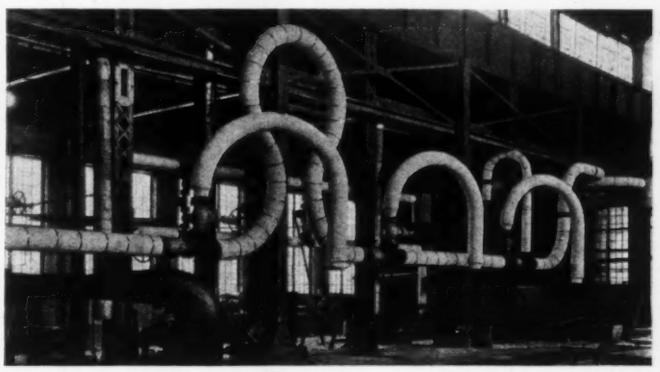
Nuts in use at present are mostly plain carbon steel either cold punched or cold pressed. In some instances they are case-hardened. These nuts to *U. S. Standard* dimensions have proved satisfactory for use up to 750 F. In a few instances the nuts have been made of the same material as the studs.

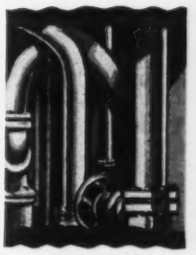
Conclusion

Materials for use at temperatures above 750 F should be divided into three classes: those to be used at 750 to 850 F, those for use between 850 and 1000 F, and those suitable for use above 1000 F. Materials to be used at these temperatures should have, at the temperature under consideration, physical properties permitting the same unit stresses as are now used in materials up to 750 F.

Pipe and fittings can be allowed a creep of 3 per cent in 200,000 hours with safety. Valve bodies and parts must be held to closer limits of total deformation because of the movable parts.

Studs of alloy steel are necessary in high temperature installations.





"Open for Discussion"

*Control of Exhaust Systems in Apartment Buildings**

A COMMON question is, "Why is there so much trouble with kitchen ventilation in apartment buildings?" Cooking smells go where they are not wanted. The exhaust may be strong, weak, or non-existent in different kitchens. Some days the exhaust is saitsfactory and others not satisfactory. Diagnosis is difficult without an inspection of the job.

Unequal temperatures, open windows, adjustment of dampers, kitchens on one side of the building, protection from the wind of certain parts of the building, some apartments rented and some not rented, maid opening the door to the back stairs, strong wind on, and whipping around the pent house; these are factors that affect performance of exhaust systems and which were discussed in Mr. Lewis' article.

Mr. Lewis suggested a method of volume control for housed fans as illustrated by Figs. 2 and 3. He says, "I then built a damper in the fan outlet as shown in Fig. 2. There was enough weight tied to the damper to hold it in almost any position desired, but the method of throttling and regulating the volume of the air delivered was unsatisfactory and irregular due to the funnel-shaped restriction which incurred the hurry-up, slide-by process. I then learned that if I turned the damper around as in Fig. 3 so that the damper opened *against*, instead of with, the air stream, I could regulate the air volume exactly and consistently."

He states further, "Whenever we have done the kitchen-exhausting by a long suction duct to a remote fan serving a number of kitchens, we have had trouble."

Mr. Lewis explains that his basic purpose in writing the original story was to develop the thought that since we cannot control the winds and cannot usually control

*This discussion is based upon an article which appeared in HEATING, PIPING AND AIR CONDITIONING for November, 1930, entitled "Handling Kitchen Exhaust in an Apartment House," by Samuel R. Lewis. The substance of this article and the diagrams which accompanied it are presented here.

the height of the buildings built by our neighbors, or the clearance between our neighbors' buildings and our own buildings, there are factors of judgment and experience which must be considered and which may go beyond the determinations of mathematics.

Comment by Prof. Lynn E. Davies[1]

I HAVE read with considerable interest the article "Handling Kitchen Exhaust in an Apartment House." I judge from the author's words that the two improvements on the single vertical duct are merely suggestions and have not actually been tried. I, therefore, feel free to suggest another method which appears feasible to me, although I frankly admit I have not actually seen it tried.

I believe the fundamental cause of trouble is the low pressures being used in these systems. The total pressure against which the fan operates consists of the resistance of the system (duct, grilles, etc.), plus the difference in pressure between the outside air and the room, due to temperature differences or any of the other causes mentioned by Mr. Lewis. This latter value may be either positive or negative. In case it is positive, by which I mean that the pressure in the room is greater than that of the outside atmosphere, the air flow from the room into the vent will be increased, whereas, when it is negative the air flow will be reduced or even reversed.

My suggestion is to introduce a high resistance of some kind between each vent and the vertical duct and then operate with a higher suction. With these higher resistances and higher suctions in the duct, the variations in pressure within the room will have very little effect upon the air flow, and if the suction maintained within the duct is larger than the largest pressure differential between room and outside atmosphere, a reversal of flow will become impossible. Furthermore, the varying heads produced by temperature differences should have very little effect on the operation of this system.

The comparative performance of the two systems may be illustrated by a comparison based on several assumed values. Suppose that a room located on an upper floor, near the fan, is subject to pressures ranging from + 0.6 to — 0.6 inches of water. Assume furthermore, that the suction at the vent is normally 0.5 inch. When the room pressure is positive the air flow will be approximately 47 per cent above normal, whereas when it is negative there will be a flow amounting to about 45 per cent of normal but in the *reversed* direction. These variations in flow may, in turn, seriously affect the pressures lower down in the riser and therefore react upon the lower rooms.

If, in this case, the duct suction is maintained at 1½ inches the same changes in room pressure will produce air flows amounting to only 118 per cent and 60 per cent of normal, respectively. This variation will not be sufficient to affect seriously the operation of the system in the other rooms. A system using a separate fan for each kitchen would be subject to at least this amount of variation unless the individual fans were selected to operate against rather higher pressures, though, of course, the most serious trouble, namely, back-drafts bringing in air from other kitchens, would be effectively prevented.

I question whether the method of using two fans, each serving half as many kitchens, would be much of an improvement over the single duct system. It is true that any abnormal condition in one kitchen could only affect four other rooms instead of a possible nine,

[1] Armour Institute of Technology, Chicago, Ill.

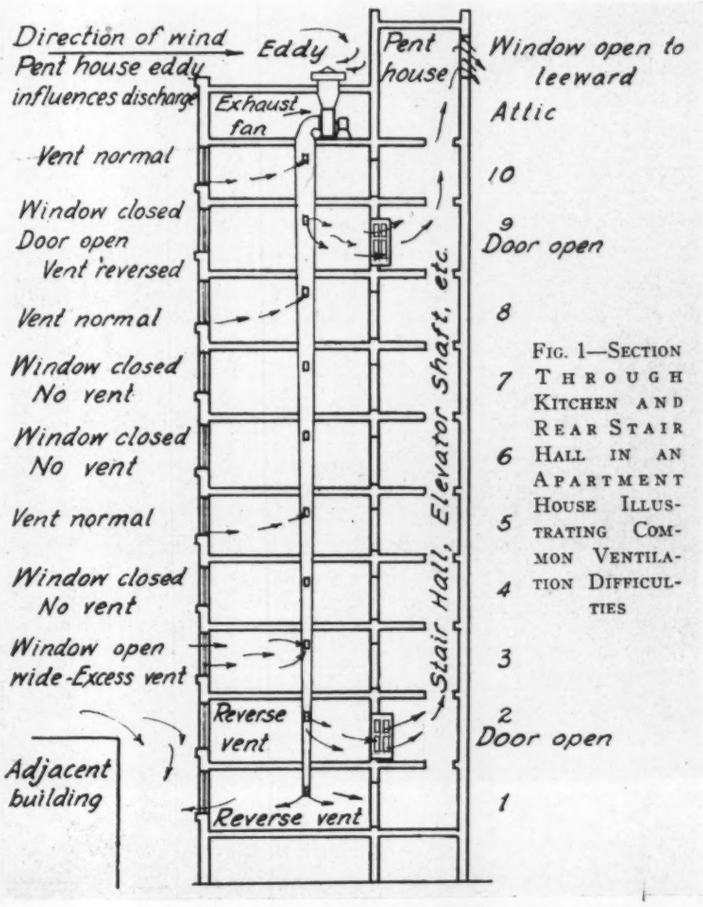


FIG. 1—SECTION THROUGH KITCHEN AND REAR STAIR HALL IN AN APARTMENT HOUSE ILLUSTRATING COMMON VENTILATION DIFFICULTIES

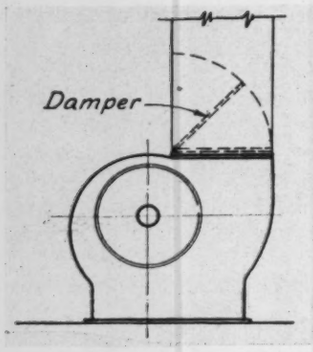


FIG. 2—INEFFECTIVE DAMPER ON A CENTRIFUGAL FAN

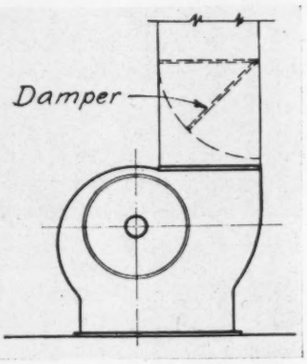


FIG. 3—AN EFFECTIVE DAMPER ON A CENTRIFUGAL FAN

but on the other hand, the effect on those four would in some cases be much more pronounced.

It is evident, of course, that with the system I suggest, any leaks in the ducts will be more serious so it will be quite essential that the risers be well constructed.

It is also evident that a larger fan will be necessary with this system, but it is my opinion that the more uniform load will permit one to be selected that will operate continuously close to its peak efficiency, whereas, the former method with its varying load makes such a selection almost impossible. For this reason, I believe, the fan size will not have to be increased as much as might first seem necessary.

I might add, in conclusion, that the method I propose has the advantage that it might, in some cases, be applied as a corrective measure to systems already installed but not functioning satisfactorily.

Mr. Lewis[2] Explains

I WILL admit that, as Professor Davies suggests, we could build a suction duct common to all of the kitchens with a fan at the top of sufficient power to maintain a stated pressure-difference around an arbitrary resistance at each inlet. I will grant that we could practicably maintain this resistance at a point where the volume of air handled would not vary materially, whether or not the kitchen windows were open and regardless of whether the corridor air pressure via the kitchen door were positive or negative and regardless of temperature.

I predict, however, that with such great resistances and necessarily such high air velocities passing the dampers* which form the resistances the kitchens, especially those nearer the fan, would be unpleasant on account of noise.

Division of the kitchen vents in a lofty building into several zones each with a separate exhaust fan, would be a benefit to the type of system suggested by Prof. Davies equally as it would be an improvement over the conventional single exhaust duct with its single low pressure exhaust fan.

[2] Consulting Engineer, Chicago, Ill.

*EDITOR'S NOTE: Prof. Davies further states: "I feel it is a mistake to think of the resistance entirely in terms of dampers or orifices. I feel that some form of resistance such as a fine mesh screen could produce the necessary results without excessive noise."

Comment by Prof. A. C. Willard[3]

I HAVE read with the greatest interest the article "Handling Kitchen Exhaust in an Apartment House." This article is written in an interesting and informative style and contains information of real value to the heating and air conditioning engineer, the architect, and the building owner.

If more people who are concerned with the construction, equipment and operation of buildings had a better knowledge of the "chimney effect" produced in many buildings whenever there is a material difference of temperatures between the inside and the outside of the building, we would have far fewer cases of complaint and dissatisfaction with the operating results obtained in such structures. An exactly similar statement applies to our lack of knowledge of "wind effect" or "air drift" across every building subjected to any wind movement whatever. When nature combines "chimney effect" and "wind effect," as so often happens in practice, it is small wonder that a system or building designed without any regard to these phenomena fails to function properly.

Mr. Lewis Replies

PROF. WILLARD comments on our lack of knowledge of wind effect or air drift across buildings. We know something about it, but we keep forgetting how to apply the knowledge. We are endeavoring to develop a code for rating unit ventilating machines. Complaint is heard frequently that many of these devices rated to warm and to deliver 1500 cubic feet of air per minute are found in practice to deliver but 1200 cubic feet of air per minute. It appears that if the unit ventilating machine has its inlet looking to windward, and there is a sizable wind pressure, the machine will deliver much more air than its rating, while if the inlet looks to leeward where there may be a decidedly negative air pressure relative to that within the building, the machine will deliver only a fraction of the air which it would deliver under normal balanced pressure conditions. One possible solution to this difficulty with unit ventilating machines is to arrange them with supply flues which come from some neutral point, wind pressure considered, and to give them enough fan power to overcome the friction of such ducts.

In order to alleviate the chimney effect in lofty build-

[3] University of Illinois, Urbana, Ill.

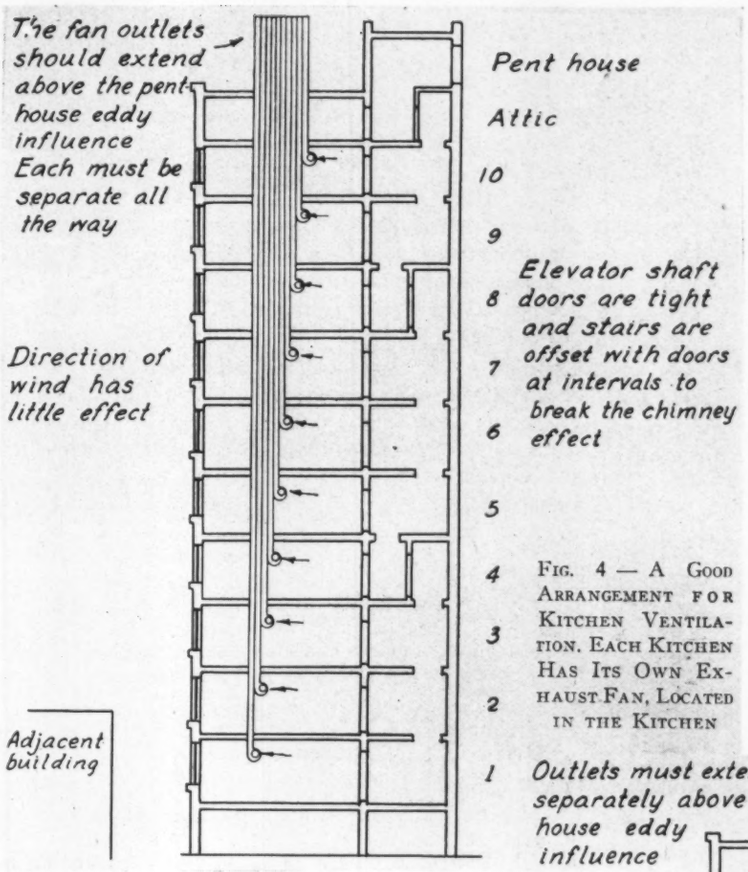


FIG. 4 — A GOOD ARRANGEMENT FOR KITCHEN VENTILATION. EACH KITCHEN HAS ITS OWN EXHAUST FAN, LOCATED IN THE KITCHEN

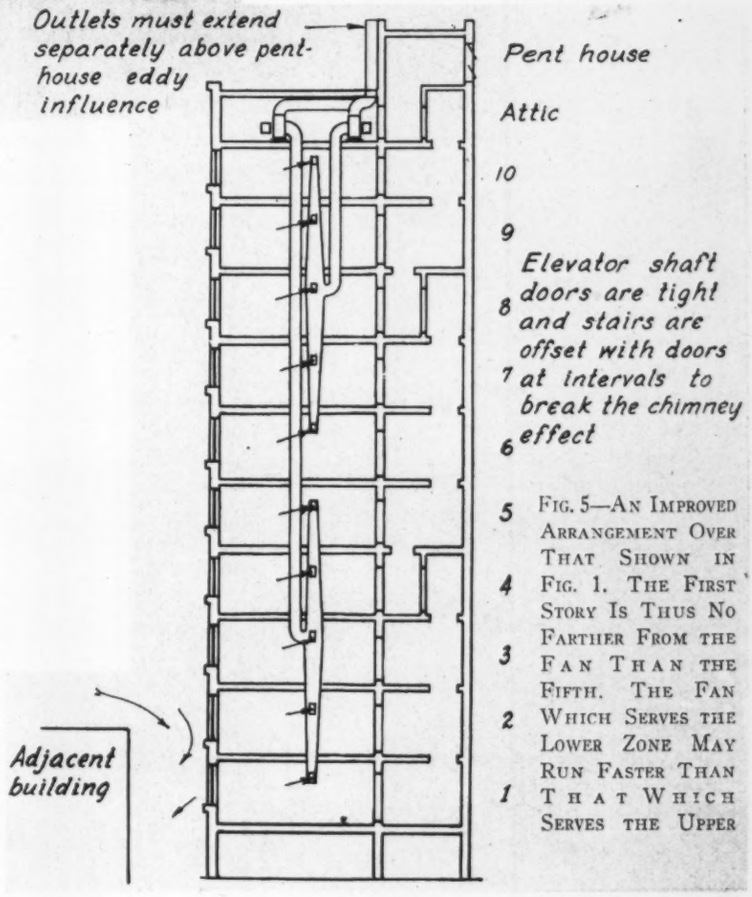


FIG. 5—AN IMPROVED ARRANGEMENT OVER THAT SHOWN IN FIG. 1. THE FIRST STORY IS THUS NO FARTHER FROM THE FAN THAN THE FIFTH. THE FAN WHICH SERVES THE LOWER ZONE MAY RUN FASTER THAN THAT WHICH SERVES THE UPPER

ings, I have in one case installed pressure fans with dust filters and tempering heaters to pump air into the stair towers from the top, creating an arbitrary pressure there to counterbalance the natural draft. This installation has proved quite pleasing in its effect and we do not find the usual hard-to-open stair doors with their whistling air leaks, their too-cold bases and too-hot tops.

In installations of gravity-indirect school ventilating systems, a successful custom was to install four large ducts for air supply, these meeting, cross-shaped, at the center of the building.

Each of the four intake openings had a no-back-draft damper, usually made of narrow horizontal strips of oiled silk. If the wind came from the north, the south intake closed automatically, and so did the east and west intakes if the north wind pressure was sufficient, and all of the indirect radiators received their air supply from windward, and there was a tendency to equalize the air pressure inside the building against that outside.

I know of a recent gesture to use this same principle for gravity cold air supply to an office building by a cross-shaped duct system at the ceiling of each story, looking out in all four directions with automatic dampers on each intake. The cool air enters the various rooms through diffusion slots close to the ceilings, where it mixes with the too-hot air in that zone.

My impression is that such a scheme as this would be far superior in general results to the common type of window ventilation, in which the windward rooms only receive outside air while the leeward rooms receive the waste and effluvia from the more fortunate windward rooms. My office looks west on the 16th story of an older Chicago office building which has open elevator shafts and stairways. When we have a west wind I am very happy. When we have a gale from Lake Michigan, however, my office, if one is air-minded and if he opens his window, is like the drainage canal, and I receive effluvia, on its way out, from the basement cigar store and restaurant and from all the seventeen stories of eastward looking tenants.

The lesson to be learned here is, of course, that elevators and stairways ought to be inclosed and as air tight as possible. The various stories must be isolated each from the others if we would achieve controlled ventilation and if we expect to achieve artificial cooling in summer.

Comments by John R. Cooper[4]

WISH to express appreciation of the courtesy extended to me in requesting that I make some comments relative to the above article. Had previously read it with considerable interest since it seemed to cover in a very careful manner the necessities of a successful installation. The manner in which the subject was covered indicates clearly why Mr. Lewis enjoys his enviable reputation.

Since the question is raised, there are, however, certain points which I wish Mr. Lewis had elaborated and, if convenient, would appreciate your requesting him to advise as to the following:

1. What difference between room and delivered air temperature was decided upon as being necessary to obtain the most satisfactory condition?
2. Presuming that the same amount of air was exhausted by the fan system as was delivered, how many re-circulating air changes per hour in each room worked out to the best advantage in practice?

These questions are raised because, as you well know, I believe that any question relative to room heating via blower equipment is a problem of air transportation.

[4] Thermal Units Manfacturing Co., New York City.

This in turn means that the governing factors are volumes delivered and exhaust air velocities, together with delivered air temperatures; dependent upon the room temperature to be maintained. Large volumes of tempered air would give uniformity of heat distribution; being limited, however, in order to avoid draft conditions.

The problem that Mr. Lewis worked out, having a combination supply and exhaust unit, must have its interesting phases and some idea as to the delivered air velocities, exhaust air velocity and volume of air handled in proportion to the size of the room would be welcome information.

Am also rather interested in finding out what the possibilities are of such a unit without danger of short circuit, although this would naturally vary with the volumes of air handled, as well as their initial and exhaust velocities.

I trust that Mr. Lewis will, at his convenience, give some information relative to the above thoughts.

Comments by Mr. Lewis

MR. COOPER'S comments seem to apply to the heating of the kitchens rather than to the removal from them of the odors. I realize, of course, that if we were able to receive a supply of benign air under a constant pressure into each kitchen, the problem of the exhaust would be very greatly simplified. This is, in the usual cooperative apartment building, rather difficult to secure.

However, if we had such an air supply and if it could be delivered toward the face of the cook at the range, I think it should never be more than eight degrees cooler than the general air of the room, no matter what the general temperature is. Too much temperature difference—no cook!

I do not understand that there is ever any question of re-circulating the air in an apartment house kitchen. Usually when meals are being prepared or while clearing up after meals the problem is one of heat removal and we would not recirculate the air at all.

One movement toward economy in operation suggests itself: Let there be a damper in the exhaust opening from the kitchen which shall be closed whenever there is no fire or heat in the range. This could be done by a simple electric switch if we used electric energy for cooking. A comparatively economical thermostatic device could be used if we burned gas or burned some other fuel.

When we are not developing local heat we are usually not developing odors, and the situation as to ventilation in the kitchen would be no worse with this automatic damper than that in the balance of the apartment. With such an arrangement we would save in fuel for heating the whole apartment, since our air supply to the kitchen to replace that withdrawn by the fan must come by induced leakage.

With such a scheme we would reduce the power cost for operating the exhaust fan, as it would handle less air, at no prejudice, while automatically we would not be ventilating the kitchen of the vacant apartment, nor that of the one of which the tenants went out to dinner!

Has the Engineer Benefitted Mankind?

●

..... is an interesting question. Most people, we think, would say "Yes." But Dr. C. E. Kenneth Mees says "No"—save in checking disease. Engineers enjoy the reputation of approaching questions with an open-mind, basing their opinions on the facts. A compilation of specific points supporting or opposing Dr. Mees' statements would, we feel, be of considerable value. There should be plenty of them.

SAVE in checking disease, science and engineering have not benefitted man, Dr. C. E. Kenneth Mees of Rochester, N. Y., Director of Research of the Eastman Kodak Company, declares in a symposium of the Engineering Foundation.

Dr. Mees doubts "if the life of the agriculturist in any country today is happier than that of a peasant in the Nile valley four thousand years ago." He questions whether the inhabitants of an American city are happier than those of a Greek or Babylonian city of the past. He would exchange the life he leads in a modern city for a life in Athens of the Periclean age or Thebes under the eighteenth dynasty.

"The one great gift of science to the world has been the diminution of disease," asserts Dr. Mees, who expresses the opinion that scientific progress will continue to accelerate, and will finally end in a revolution in our whole social and economic life.

Dr. Mees' statement followed a declaration by the presidents of the four societies of civil, mining, mechanical, and electrical engineers, that the machine age had promoted human happiness.

"In considering the fundamental problem of this symposium, namely, whether engineering progress — under which term, presumably, the engineer will modestly include the development of science and the arts through the ages—has, on the whole, been a benefit to mankind, we must first define what is meant by 'benefit,' and this involves a consideration of the aim or purpose which may be ascribed to man's life on the earth," Dr. Mees says.

"The aim which will be ascribed to man's life will depend upon the person who ascribes it. The biologist would probably say that that purpose was the reproduction of the species. The mystic would say that the chief aim of man was to worship and glorify his maker; the materialist would say that it was to achieve happiness, while the Greek philosopher would say that it was to increase his knowledge of good. In considering the effect of scientific progress through the ages, therefore, we must consider its effect upon each of these aims as well as upon any others which may occur to us.

"If the purpose of the race is to multiply the species, then engineering progress has certainly been successful, since the population of the world is increasing rapidly and promises to become a very great problem in the near future.

"The advance of civilization seems to offer no consolation to the mystic. If men are more ethical than they used to be, they are certainly becoming less religious, and whatever future developments may show, at the present time engineering can claim no triumphs in the field of religion.

"This leaves us with the last two categories: the acquisition of happiness by the individual and his progress towards a knowledge of good. It is under these two headings that my colleagues would think that engineering progress had been of benefit to mankind, including under the heading of 'happiness' all the material benefits which it has conferred, and under the heading of 'good' all the cultural and intellectual benefits.

"Happiness, however, is not a thing which depends upon the possession of material things beyond a certain minimum. Happiness depends only on the spirit provided that the spirit is not oppressed by the material situation of the body; it is fundamentally a by-product of activity of some kind. A man can be happy in arduous toil or in nerve-racking sport; he can be happy in intellectual relaxation; he can be happy, passively happy

at any rate, in mere contemplation. Have we not all heard of contented cows?

"The one great gift of science to the world has been the diminution of disease. The diseases which are due to micro-organisms, especially those which afflict the young, are diminishing very rapidly as a result of the exploration of their nature and an application of the discoveries which have been made. The diseases due to deficiency are in their turn being diminished, and will eventually be destroyed. We may look forward without doubt to a world in which widespread pestilences can exist no more. This is certainly a benefit to man which may be claimed by engineering.

"But, apart from this, I doubt if the life of the agriculturist in any country today is happier than that of a peasant in the Nile valley four thousand years ago. It is true that in the more advanced countries he has had great aid in his struggle with the pests that afflict his crops and that in many cases he has had irrigation supplied as the result of applied engineering, the first engineers to do this having lived more than four thousand years ago.

"Moreover, in a few countries, and especially in the United States, machinery has diminished the heavy labor of the old time farmer, and has added many comforts to his domestic life. But, while machinery assisted the farmer at first, it is now undermining his status, and may perhaps end by eliminating him, so that the old normal life of man, in which he cultivated the soil and lived on the crops, exchanging a small surplus for goods produced by somebody else, is passing possibly never to return.

"Engineering can produce that surplus and supply it to the consumer at a far lower labor cost than can the individual producer, and throughout the world economists and statesmen are considering the future of the farmer with troubled faces, remembering that farming is the normal life of man, and that any other life for the mass of mankind involves the greatest readjustment in our whole economic and social system.

"To turn from the agriculturist to the town dweller and even to the intellectual of the city, will any student of history agree that the inhabitants of an American city are on the whole happier than those of a Greek or a Babylonian city of the past? This is a thing difficult to assess, but for myself if I could exchange the life that I lead in a modern city for a life in Athens of the Periclean age or Thebes under the eighteenth dynasty, I think that I should undoubtedly accept it.

"In those days there was more leisure, less pressure, more opportunity for the exchange of ideas, less emphasis on material things. There is little that a man can get today which he could not have had in Athens. It is better to walk a short distance than ride a long one. It is better to talk to one's friends and to attend in a leisurely manner to one's business than to dash off to the office, returning tired in the evening. The life in a small city in a house with a garden is a better life, it seems to me, than a life in a New York apartment.

"It may perhaps be claimed that the urban worker has gained greatly in comfort by the progress of engineering and that in recent years, at any rate, life in the sordid slums of our great cities has been mitigated by technical progress.

"It may be true that a factory operative in a modern American textile mill has a much greater chance of happiness than a corresponding operative in an English mill town of seventy years ago, but it seems to me that the nadir of human existence was reached under those factory conditions of the first half of the nineteenth century and that the lot of the modern worker should be compared more properly with the poorer classes of the great cities of the ancient world, before the machines of the engineer had begun to replace the handwork of the craftsman. And I feel that a freeman of the Roman Empire, an apprentice of the middle ages, and perhaps even a Grecian helot, had at least as good a chance for happiness as that of most modern factory workers.

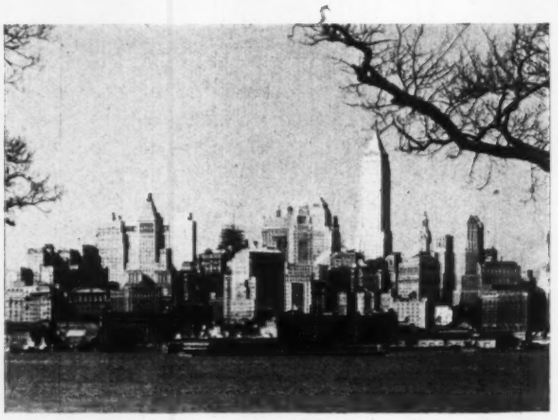
"When we turn to the last aim of mankind, that which we may define as progress in culture, it seems to me that the modern world is a very much less satisfactory one than the world of the past. Culture demands primarily leisure, and at the present moment, at any rate, leisure is conspicuously lacking in the world. It demands intellectual detachment, and intellectual detachment is exceedingly hard to come by.

"Some of my colleagues may urge that engineering progress has contributed to international understanding. To this I would enter a definite objection. I doubt if nations have ever hated each other as fervently as they do at the present moment. Looking back on history, one can watch the waxing and waning of international good feeling. At the present moment it seems to me it is nearly a minimum.

"What then of the future? Shall we stop scientific progress and shall we try to get back to the world of the past? To this I would say that it is of no use to discuss it. Scientific progress will continue; it will continue to accelerate. In my opinion that acceleration will finally end in a revolution in our whole social and economic life.

"What form that revolution will take and what type of life will come out of it, we cannot tell. I believe, however, that in some respects the life of the future will be closer to the life of the past than it will be to the life of today. I believe that a large portion of mankind will abandon the feverish quest for material things, and will employ its greater leisure in the development of art and in the cultivation of its soul.

"This Utopia of the future may be possible primarily because of engineering progress, and in that sense at some time engineering progress may be a blessing to mankind. Up to the present time I doubt whether on the whole it has been."



Uses of

Air Conditioning Equipment

in the

Manufacture of Gas

By Malcolm Tomlinson*

MANUFACTURED gas consists of a number of gases such as hydrogen, carbon monoxide, methane, benzine and naphthalene. It is usually produced by one of two processes; one involves the use of soft coal alone and is thus termed coal gas. The other uses both soft coal and crude oil and is known as water gas because steam, in this process, is brought into intimate contact with incandescent carbon. In the soft coal regions there is still another process for manufacturing gas through by-product coke ovens.

Contain Hydrogen Sulphide

All of these gases contain hydrogen sulphide after they have passed through the scrubbing and condensing equipment. Since hydrogen sulphide in the presence of moisture (H_2O) forms sulphuric acid in minute quantities it must be eliminated. In the atmosphere it would irritate the nose, throat, lungs and eyes. In machinery, such as gas engines, it would corrode metals. Its removal is accomplished in boxes containing iron oxide, which are known as purifiers, through which the manufactured gas is forced to pass before it reaches the gas holder.

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Reaction Depends on Presence of Water

The reaction necessary for the removal of the hydrogen sulphide (H_2S) depends on the presence of water. In other words there could be no removal if the manufactured gas and the iron oxide were both dry. Therefore water must be present in either the gas or the oxide. Although the effect of moisture in the gas has long been realized the method employed commercially has been to moisten the oxide by means of saturated or exhaust steam.

The amount of moisture required depends to a considerable extent on various compounds of iron, or ferric, oxide present. Of these the most active for the purpose under discussion is ferric hydroxide or ferric oxide hydrated.

Two reactions must be taken in consideration in the removal of hydrogen sulphide from manufactured gas. One is the sulphiding process in which ferric sulphide is formed with the aid of water, hydrogen sulphide and ferric oxide. The other is the revivifying process in which the oxygen of the air reacts to form the iron hydrate and sulphur. These reactions may be expressed as follows:

Sulphiding process—

$$\underset{\text{Iron Oxide}}{Fe_2O_3 . x\,H_2O} + \underset{\text{Water}}{} \quad \underset{\text{Hydrogen Sulphide}}{3H_2S} = \underset{\text{Ferric Sulphide}}{Fe_2S_3} + (x+3)\,\underset{\text{Water}}{H_2O}$$

Revivifying process—

$$\underset{\text{Ferric Sulphide}}{2Fe_2S_3} + \underset{\text{Oxygen}}{3O_2} + \underset{\text{Water}}{2vH_2O} = \underset{\text{Ferric Oxide}}{2Fe_2O_3.yH_2O} + \underset{\text{Water}}{} + \underset{\text{Sulphur}}{6S}$$

In the first equation the iron oxide is mixed with water or the hydrogen sulphide carries water into the mixture. In the second water enters the process not only with the oxygen from the air but also with the ferric sulphide. As we have seen the actual process of sulphiding consists of introducing steam and of producing water. In the revivifying process water is absorbed. From the equations presented it will be seen that the amount of moisture required for the first process is not necessarily the same as for the second process.

Correct Humidity of Gas Stream Determined

For a long time the problem of determining the correct moment for turning on the steam for the iron oxide

boxes and also of turning it off was acute. Finally, in 1921, a rather clever solution was found. Litmus paper has the peculiar property of turning to a red color in the presence of an acid and to a blue in the presence of an alkali. As hydrogen sulphide, in the presence of water, reacts to form sulphuric acid it became evident that litmus paper could be used as a detector of the strength of the hydrogen sulphide in the gas. This detector was adopted quite generally but even this improvement was not sufficient to give a satisfactory control. At last, in 1930, the *American Gas Association* and *Johns Hopkins University* cooperated in an investigation to determine the correct humidity of the gas stream for maximum efficiency in hydrogen sulphide removal.

The Tests

In the sulphiding tests the temperature was maintained at 100.4 F. Under this condition the removal of hydrogen sulphide increased rapidly as the relative humidity was raised until a relative humidity of 65 per cent was reached. Above this humidity the efficiency of removal fell off rapidly. Since no data were obtained at other temperatures it is not possible to state that 65 per cent is the optimum humidity. Nevertheless the important point to be observed is that best sulphiding was not had at saturation. Another interesting observation is that 65 per cent relative humidity has, in a remarkable number of cases in many diverse industries, proven to be the optimum humidity after research has brought out the facts.

Since very little was known regarding the reactions in the revivifying process a more complete investigation was undertaken. The rate of revivification was apparently not affected by the oxygen content of the air and the rate of flow of the gas. As far as temperatures were concerned it was found that 100.4 F was the most suitable.

A particular example of the temperature effect might be of interest in this connection. "At 68 per cent relative humidity the oxide revivified more rapidly at 74.3 F than it did at humidities below 20 per cent at 100.4 F." The effect of relative humidity was quite different than for the sulphiding process. A humidity of 10 per cent had no effect over a period of one year. At 20 per cent some revivification had occurred in 5,780 hours. At 35 per cent humidity revivification took place in 18 hours, at 65 per cent it took only 2¾ hours while at 100 per cent revivification was practically instantaneous. At this latter humidity water is likely to condense during the reaction and this alone would make the use of saturation inadvisable. Best results can, therefore, be had at high humidities somewhat below saturation.

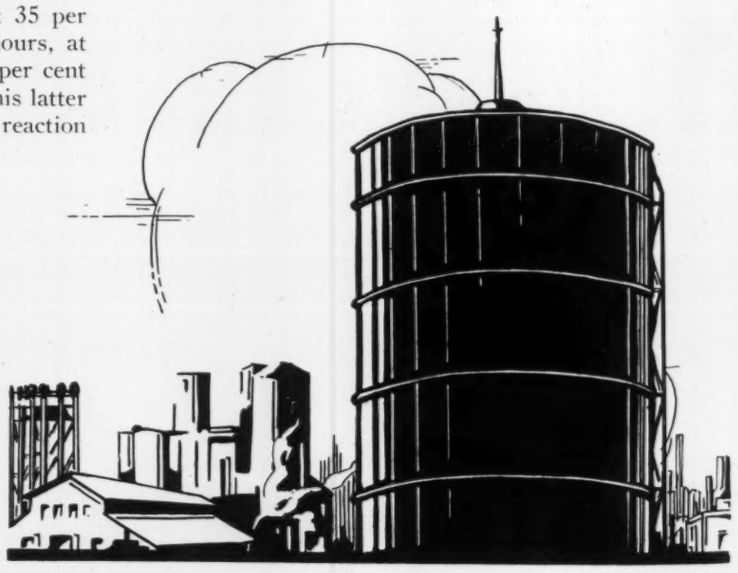
Effect of Control of Humidity

Tests run to determine the effect of control of the humidity of the gas as against no control were positive in their results. At no time did the uncontrolled purifier give results as good as the controlled purifier when the same iron oxides were used. The most interesting phase of this portion of the investigation developed was the fact that a fouled purifier could become highly efficient when the best relative humidity for the gas was employed. This latter fact indicates that humidity control will decrease markedly the cost of replacements for purifiers.

Various methods for the control of the relative humidity were suggested as a result of the investigation. Where steam is used it is suggested that a regulated amount of live steam be introduced with the exhaust steam in order to cut down moisture entrainment. Sprays throwing a fine mist into the gas ahead of the purifiers was also suggested. Finally an "automatic mechanical means for suitably controlling the temperature and the humidites of the gas" was suggested.

This latter suggestion should be of interest to the air conditioning manufacturers. It should be doubly interesting to the gas manufacturers since it solves entirely the problem of moisture entrainment which will always be present where steam or water sprays alone are employed.

The large producers of manufactured gas are intensely interested in the results of these investigations. While new natural gas wells are coming into existence constantly the main supply of gas today is that produced by the coal and water gas processes. The gas industry which, in 1921, seemed to be on a rapid decline, is growing rapidly in many directions. In this new gas era those methods which will eliminate trouble and cut down replacement costs are bound to receive attention so that it seems certain that this new field, which is now opening for air conditioning equipment, will be worth watching.



Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

AUGUST 1931

Material in this Section prepared by Publication Committee: W. A. Rowe, *Chairman;* E. O. Eastwood, and F. D. Mensing A. V. Hutchinson, Manager of Publications

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Using Gas-Vapor Mixtures for Heating Purposes

By C. A. Dunham[1], Chicago, Ill.
MEMBER

THIS paper is the result of original research on the use of gas-vapor mixtures for heating purposes. The principle involved is that of burning gas in the presence of water and at pressures slightly below atmospheric, using a mixture of the combustion products and steam for conveying heat to the *radiators*.

Description of Apparatus

An especially designed generator with suitable gas controls was employed (see Figs. 1 and 2). The refrac-

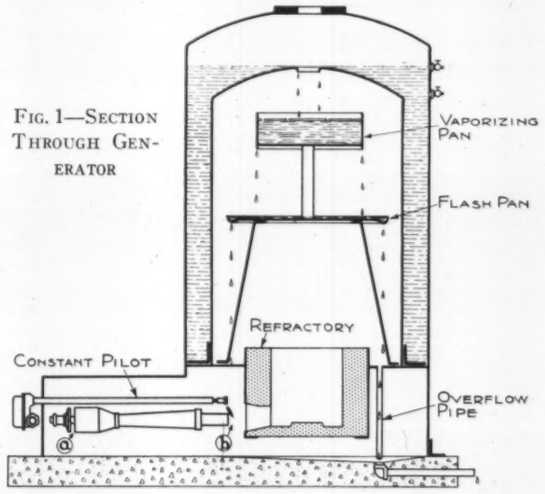


FIG. 1—SECTION THROUGH GENERATOR

tory with circular interior receives the gas flow introduced on one side near the bottom. The flame travel within this bowl is increased by means of the whirling action imparted by the rapid expansion of the burning gases injected into the refractory. The primary air is drawn into the burner at *a* by the natural inspirating action of the gas flow, and the secondary air is drawn in between the end of the burner and the refractory opening at *b* due to the lower pressure within the refractory. This air and gas mixture ignites off the end of the burner by means of a constant burning pilot and the flame passes through the hole into the refractory which soon becomes heated, and perfect combustion takes place before the flame reaches any cooler surfaces.

It will be observed in Fig. 1 that the generator has a double wall and that the dome of the inner wall is open at a point directly below the supply outlet of the generator. When the water level between the walls rises to an overflowing point (due to the expansion by heat) the water drips into a pan located directly below the opening, as indicated. This pan in turn overflows on to the flash plate which is slightly cupped to hold a shallow supply of water. Any surplus water passes to waste by dripping off the edge and down into the drain through the overflow pipe.

A complete system as it was applied to a house is shown diagrammatically in Fig. 3. A one-tenth horsepower electric motor operates an exhauster which in turn exerts a suction through the *radiators* and generator. It starts operating when the room push button is pressed and simultaneously brings on the gas for ignition by the constant burning pilot. This push button may be further supplemented to operate under the control of a room thermostat.

After the gas is ignited, the sheet of water reaches the flash pan and is giving off vapor with no appreciable lapse of time. The products of combustion, after mixing with the vapor, are drawn immediately into the *radiators* where the vapor is condensed, the gases cooled, and the heat thus passed to the room being heated. The condenser vapor is returned to the water space between the walls of the generator.

The body of water which lies back of the inner wall of the generator and in direct contact with it, responds to the initial heat input by expanding, thus causing a trickling or overflowing into the vaporizing pan and then dripping to the flash pan below. This flow is made continuous by the constant return of the condensed vapors. These heating surfaces coupled with the surface area of the drops of water, present a large heating area and cause surprising results in vaporizing possibilities.

The question might arise as to why water is used in-

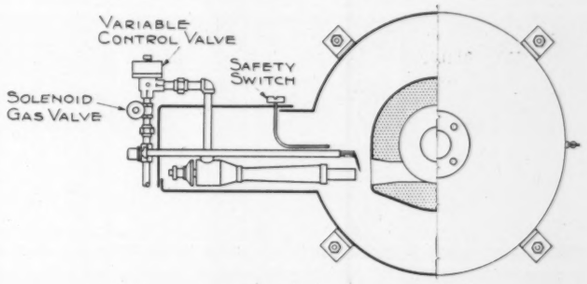


FIG. 2—TOP VIEW OF GENERATOR

stead of the gases to carry heat. The primary reason is that gas is not an economical heat vehicle. Without the water content a very high temperature would be created within the generator. The gases which are carried to the heat dissipating medium then bear no established relation in temperature to those in the generator,

[1] President, C. A. Dunham Co., Chicago, Ill.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1931.

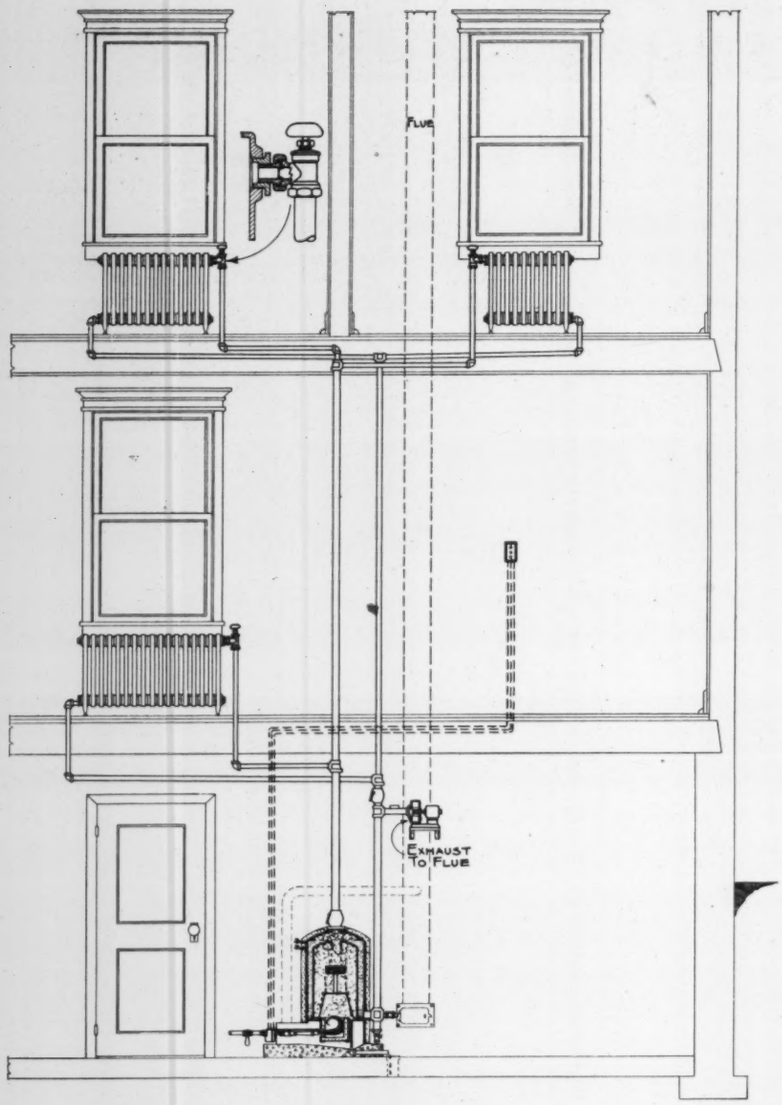


Fig. 3—Complete System Applied to a House

and the distance of the travel of gases at given temperatures through the piping is very short.

A test without any surplus water content showed a temperature of 600 F at the outlet of the generator. At the entrance to the first *radiator* 20 ft from the generator, the temperature was only 90 F, and the *radiator* would not heat completely without undue forcing.

By adding water, the conducting capacity of the pipes was materially increased and the temperature of the generator decreased to a level corresponding more closely to the heat input to the farthest *radiator*. In other words, an appreciable portion of the sensible heat in the gases was absorbed as latent heat and conveyed to the most distant points by the better heat carrying medium created by the additional water content.

In operation, practically all vapor returns to the generator after having given up its heat by condensing. The hydrogen content of the gas by combination with oxygen through combustion processes provides a surplus of water beyond that required by the generator to form the heating medium and above the slight losses through the exhausted gases; this surplus water passes away through the drain. All noncondensable gases when cooled are discharged to the flue by the exhauster. A recent test was made to obtain the efficiency of the generator and also a heat balance to account for all the heat losses of such a type of heating system.

Test Procedure

Fig. 4 is a diagram of the complete test set-up. Figs. 5 and 6 show end and side views of the apparatus. A unit heater was installed in the regular floor stand with the outlet connected to a duct approximately one foot square. As shown in these figures the generator, unit heater and a portion of the duct were insulated.

Temperatures were taken by means of thermocouples in conjunction with the Leeds and Northrup type K potentiometer. Where extreme accuracy was not essential, temperatures were taken into calibrated glass stem thermometers. The flue gases were analyzed frequently throughout the test by means of the improved Hays apparatus.

In order to insure uniform temperatures the apparatus was operated continuously for a period of several hours

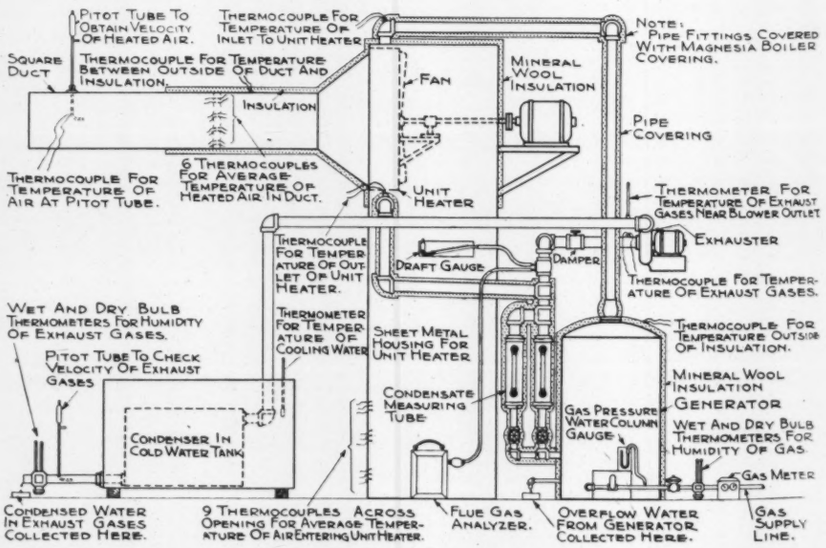

Pitot Tube To Obtain Velocity Of Heated Air.
Thermocouple For Temperature Of Inlet To Unit Heater
Square Duct
Thermocouple For Temperature Between Outside Of Duct And Insulation.
Insulation
Fan
Mineral Wool Insulation
Note: Pipe Fittings Covered With Magnesia Boiler Covering.
Pipe Covering
Thermocouple For Temperature Of Air At Pitot Tube.
6 Thermocouples For Average Temperature Of Heated Air In Duct.
Unit Heater
Thermometer For Temperature Of Exhaust Gases Near Blower Outlet
Exhauster
Thermocouple For Temperature Of Outlet Of Unit Heater.
Draft Gauge
Damper
Thermocouple For Temperature Of Exhaust Gases.
Wet And Dry Bulb Thermometers For Humidity Of Exhaust Gases.
Thermometer For Temperature Of Cooling Water
Sheet Metal Housing For Unit Heater
Thermocouple For Temperature Outside Of Insulation.
Pitot Tube To Check Velocity Of Exhaust Gases
Mineral Wool Insulation
Condensate Measuring Tube
Generator
Gas Pressure Water Column Gauge
Wet And Dry Bulb Thermometers For Humidity Of Gas.
Condenser In Cold Water Tank
Gas Meter
Condensed Water In Exhaust Gases Collected Here.
9 Thermocouples Across Opening For Average Temperature Of Air Entering Unit Heater.
Flue Gas Analyzer.
Overflow Water From Generator Collected Here.
Gas Supply Line.

Fig. 4—Complete Test Set-Up

FIG. 5—END VIEW OF TEST APPARATUS

before beginning the test. The readings were recorded for two more hours, an average of five readings, one each half hour, being used as the basis for the results of this report.

Data and Calculations

(1) *Heat Value of Gas.* The heat value of the gas was obtained from an analysis made by the Northern Indiana Public Service Company, the net heat content averaging 577 Btu per cubic foot. The composition of the gas was as follows:

CO_2	= 1.8 per cent
O_2	= 2.8
N_2	= 16.0
CO	= 6.0
H_2	= 33.2
CH_4	= 35.1
C_2H_6	= 1.5
Illuminants	= 3.6
Total	= 100 per cent

The gross heat value on this basis was 641.1 Btu per cubic foot.

(2) *Correction of gas as metered to standard conditions of 60 F, 30 in. mercury and 0 per cent humidity.*

Conditions of gas at meter during the test:

Average wet-bulb temperature	55.1 F
Average dry-bulb temperature	71.6 F
Pressure above atmosphere	4¾ in. water
Atmospheric pressure	29.50 in. mercury
Absolute pressure of the gas	29.85 in. mercury

Consider vapor pressure in the gas equal to vapor pressure in air having the same wet-bulb and dry-bulb temperatures = 0.3 in. mercury. Absolute pressure of dry gas as metered = 29.85 — 0.3 = 29.55 in. mercury. Correction factor for converting cubic feet of gas as metered to dry gas at 60 F and 30 in. mercury = $\frac{460+60}{460+71.6} \times \frac{29.55}{30} = 0.9635$. Average gas consumption as metered during test = 124.4 cu ft per hour. Dry gas at standard conditions = 124.4 × 0.9635 = 119.9 cu ft per hour.

(3) *Water vapor supplied to the system during the test.*

(*a*) By gas supply

Condition of gas entering burner: Wet-bulb temperature, 55.1 F; dry-bulb temperature, 71.6 F.

Consider water content of gas equal to that of air at same wet- and dry-bulb temperatures = 0.0004 lb per cubic foot of gas as metered. Weight of water in gas supply = 124.4 × 0.0004 = 0.050 lb per hour.

(*b*) By air supply

Conditions of air used during test: 71.2 F; 29.50 in. mercury; 38 per cent relative humidity.

Saturated vapor pressure at 71.2 F = 0.7695 in. mercury. Density saturated vapor pressure at 71.2 F = 0.001201. Vapor pressure in air as used = 0.38 × 0.7695 = 0.292. Pressure of dry air = 29.50 — 0.29 = 29.21.

Volume of dry air at 60 F and 30 in. mercury used per hour = 808.2 cu ft. [See item (5).] Volume of air as used = $\frac{460+71.2}{460+60} \times \frac{30}{29.21} \times 808.2 = 847.9$ cu ft per hour. Moisture content of air as used = 0.38 × 0.001201 = 0.0004564 lb per cubic foot. Weight of water in air supply = 847.9 × 0.0004564 = 0.387 lb water per hour.

Total water vapor supplied to system from moisture content in the air and by the gas supply = 0.050 + 0.387 = 0.437 per hour.

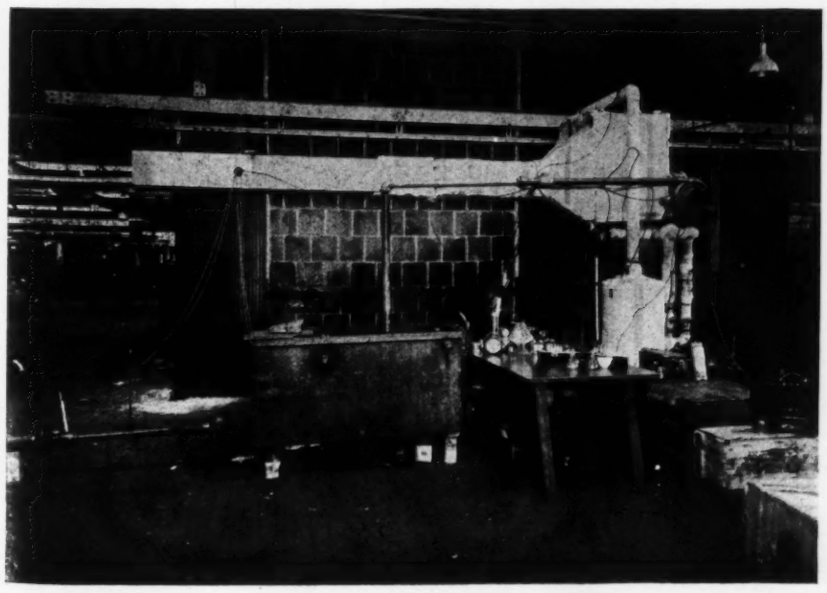


FIG. 6—SIDE VIEW OF TEST APPARATUS

(4) *Water produced by combustion.* A vacuum of one inch of water existed at the top of the generator connection. The combustion of one cubic foot of the dry gas at 60 F and 30 in. mercury produces 1.185 cu ft of water at the same conditions. The theoretical volume if water vapor were a perfect gas at standard conditions = $1.185 \times 119.9 = 142.1$ cu ft of H_2O produced per hour. Weight of water produced by combustion = $142.1 \times 0.04758 = 6.76$ lb per hour.

(5) *Per cent excess air.*

Average analysis of flue gas:

CO_2	=	8.7 per cent
O_2	=	3.7
CO	=	0
N_2	=	87.6
Total	=	100.0 per cent

Excess O_2 per cubic foot of flue gas = 0.037 cu ft. Necessary O_2 per cubic foot of flue gas = $0.226 - 0.037 = 0.189$ cu ft.

Per cent excess O_2 and air = $\frac{0.037}{0.189} = 19.6$ per cent. Air required for combustion of one cubic foot of gas plus excess air = $5.633 \times 1.196 = 6.741$ cu ft. Dry air supplied = $6.741 \times 119.9 = 808.2$ cu ft per hour, at standard conditions.

(6) *Non-condensable flue gases per hour.* Total volume of non-condensable flue gases per cubic foot of dry gas burned in test equals volume of gases produced by combustion plus excess air = $5.308 + 5.633 \times 0.196 = 6.418$ cu ft. Non-condensable flue gases per hour = $6.418 \times 119.9 = 769.5$ cu ft.

(7) *Water vapor which would have escaped per hour during test if gas had burned at an efficiency of 100 per cent.* If gas were burned in air at an efficiency of 100 per cent under standard conditions, perfect combustion would be obtained without excess air, and the products of combustion would be cooled to 60 F at 30 in. mercury pressure. In other words, the gross heat value of the gas would be obtained. The volume of non-condensable flue gases per cubic foot of dry gas burned at 100 per cent efficiency = 5.308 cu ft. (Gas volume at standard condition.) Gases remain saturated at 60 F. Saturated vapor pressure at 60 F = 0.5214 in. mercury. Pressure of dry gases = $30.0 - 0.52 = 29.48$ in. mercury. Volume of saturated gases = $\frac{30}{29.48} \times 5.308 = 5.402$ cu ft. Density of saturated vapor at 60 F = 0.000829 lb per cubic foot. Water vapor escaping per cubic foot of gas burned = $5.402 \times 0.000829 = 0.004479$ lb. Water vapor which would have escaped during test if gas had burned at an efficiency of 100 per cent = $119.9 \times 0.004479 = 0.54$ lb per hour.

(8) *Water carried to atmosphere by flue gases.* Non-condensable flue gases per hour at standard conditions [from item (6)] = 769.5 cu ft. Gases enter the amtosphere saturated at 93.6 F. Atmospheric pressure = 29.50 in. Saturated vapor pressure at 93.6 F = 1.589 in. Pressure of dry gases = $29.50 - 1.59 = 27.91$ in. mercury. Volume of gases entering atmosphere = $\frac{460 + 93.6}{460 + 60} \times \frac{30}{27.91} \times 769.5 = 880.6$ cu ft per hour. Vapor content = 0.002375 lb per cubic foot. Water vapor carried to atmosphere = $880.6 \times 0.002375 = 2.09$ lb per hour.

(9) *Heat loss of system through liquid and vapor loss.* This is equal to the water whose heat of liquid above 72 F and heat of vaporization is lost.

Water collected at condenser outlet plus water escaping to gas minus water vapor which would have escaped through flue atmosphere minus water vapor supplied to system by air and if gas had burned at 100 per cent efficiency = $4.40 + 2.09 - 0.05 - 0.387 - 0.54 = 5.51$ lb per hour. This entered the exhauster as vapor at a temperature of 130 F. Total heat of vaporization lost = $5.51 \times 1{,}020 = 5{,}620$ Btu per hour. Heat of liquid lost = $5.51 \times (130 - 72) = 320$ Btu per hour. Heat of liquid of overflow from generator is lost = $0.47\ (150 - 72) = 36$ Btu per hour. Total heat lost as heat of vaporization and heat of liquid = $5{,}620 + 320 + 36 = 5{,}976$ Btu per hour.

(10) *Heat lost from system as sensible heat of dry gases.* Heat lost from system as sensible heat of dry gases based on flue gas analysis = $880.6 \times 0.06850 \times 0.24 \times (130 - 72) = 839.7$ Btu per hour.

(11) *Total accountable heat loss.*

Latent heat	= 5,620 Btu per hour	[Item (9)]
Heat of liquid	= 320 Btu per hour	[Item (9)]
Sensible heat of gases	= 840 Btu per hour	[Item (10)]
Heat loss through flue	= 6,780 Btu per hour	

Total accountable heat loss = $6{,}780 + 36$ Btu at overflow [Item (9)] = 6,816 Btu per hour.

(12) *Heat produced by combustion of flue gas.* Gross heat value of fuel gas burned per hour = $119.9 \times 641.1 = 76{,}841.6$ Btu [See items (1) and (2).]

(13) *Velocity of flue gas and vapor through 2-in. pipe discharge to atmosphere.*

Condition of gases at point velocity head was measured: saturated at $t = 93.6$ F and $b = 29.50$ in.

$V = k\sqrt{h}$

$k = 952.8 \times \sqrt{\frac{460 + 93.6}{29.50}} = 4{,}125.6$

Average $\sqrt{h} = 0.212$ (h measured at center of pipe)

Correction for $\sqrt{h}$ taken as 0.933

Correction for density = $\sqrt{\frac{0.07068}{0.07088}} = \sqrt{0.9972} = 0.9986$

$V = 4{,}125.6 \times 0.175 \times 0.933 \times 0.9986 = 676.5$ ft per minute.

(14) *Rate of flue gas discharge to atmosphere* = $Q = A \times V$. A = 0.0233 sq ft (for 2-in. pipe). $V = 676.5$ ft per minute [from item (13)]

$Q = 0.0233 \times 676.5 = 15.76$ cu ft per minute or 945.6 cu ft per hour.

(15) *Density of flue gases.* Density of dry flue gases at 93.6 F and 29.50 in. = $1.0244 \times 0.07068 = 0.07240$ lb per cubic foot. Pressure of saturated vapor at 93.6 F = 1.589 in. of mercury. Density of saturated vapor at 93.6 F = 0.002375 lb per cubic foot. Pressure of dry gases in mixture = $29.50 - 1.59 = 27.91$ in. of mercury. Density of dry gases at 93.6 F and 27.91 in. mercury = $\frac{27.91}{29.50} \times 0.07240 = 0.06850$ lb per cubic foot. Weight of one cubic foot of the gas vapor mixture = $0.06850 + 0.00238 = 0.07088$ lb.

(16) *Water carried to atmosphere by flue gases.* Gases and vapor discharged at 93.6 F and 29.50 in. mercury = 945.6 cu ft per hour. The water vapor per cubic foot of mixture = 0.002375 [item (15)]. Water carried to atmosphere by flue gases = $945.6 \times 0.002375 = 2.25$ lb per hour.

(17) *Total heat of vaporization lost.* By procedure similar to

Symbols Used in This Paper

A = internal area of pipe or duct, square feet
b = absolute pressure, inches of mercury
h = velocity head, inches of water
k = a constant
Q = flow or discharge, cubic feet per minute
t = temperature of air, degrees Fahrenheit
V = velocity, feet per minute
W = flow or discharge, pounds per hour

that given in [item (9)], the total heat of vaporization lost = 5783 Btu per hour, and the heat of liquid = 329 Btu per hour lost in the flue gases.

(18) *Heat lost as sensible heat of non-condensable flue gases* = $945.6 \times 0.06850 \times 0.24 \times (130 - 72) = 901.7$ Btu per hour.

(19) *Heat loss through flue and from overflow.*

Latent heat	= 5783 Btu per hour
Heat of liquid	= 329
Sensible heat of gases	= 902
Total loss through flue	= 7014 Btu per hour
Loss from overflow	= 36
Flue and overflow losses	= 7050 Btu per hour

(20) *Velocity of air in discharge duct* $= V = k\sqrt{h}$

For dry air $k = 952.8\sqrt{\dfrac{460 + t}{b}}$

Condition of air in duct during test at point velocity head was measured: Average $t = 97.6$ F; $b = 29.50$ in; relative humidity = 12½ per cent.

$$k = 952.8\sqrt{\frac{460 + 97.6}{29.5}} = 4134.6$$

Average $\sqrt{h} = 0.648$ (h taken at center of duct). Correction factor for $\sqrt{h} = 0.933$. Density of dry air at 97.6 F and 29.50 in. mercury = 0.070186 lb per cubic foot. Density of air at 12½ per cent relative humidity, 97.6 F and 29.50 in. mercury = 0.069986 lb per cubic foot.

$$\text{Correction factor for humidity of test air} = \sqrt{\frac{0.070186}{0.069986}} = 1.0014.$$

$V = 4{,}134.6 \times 0.648 \times 0.933 \times 1.001 = 2{,}502.0$ ft per minute.

(21) *Rate of air flow through discharge duct* $= Q = A \times V$. A (of duct) = 0.989 sq. ft. V [from item (20)] = 2502.0 ft per minute. $Q = 0.989 \times 2502 = 2474.4$ cu ft per minute, or 148,464 cu ft per hour. Density of air under test conditions (12 per cent relative humidity, 97.6 F, 29.5 in. mercury) = 0.069986 lb per cubic foot. $W = 148{,}464 \times 0.069986 = 10{,}390$ lb per hour.

(22) *Heat delivered by radiator* equals pounds of air per hour times average specific heat times temperature rise in degrees Fahrenheit. Average specific heat of test air = 0.243. By test it was found that the air was heated 1.5 deg by the action of the fan. Thus the actual rise of temperature due to generator = 101.1 — (72.7 + 1.5) = 26.9 deg. Heat delivered by radiator = $10{,}390 \times 0.2430 \times 26.9 = 67{,}920$ Btu per hour.

(23) *Heat balance*

Heat delivered by *radiator*		67,920 Btu per hour
Lost through the flue		
(*a*) by flue gas analysis	6780	
(*b*) by Pitot tube measurements	7014	
Average		6,897
Lost in overflow		36
Unmeasured losses, including heat loss through insulation		1,989
Total heat output of unit		76,842 Btu per hour
Total heat input to unit		76,842 Btu per hour

(24) *Calculation of total heat loss through flue if flue gases had been cooled to 97 F at the exhauster:* Non-condensable flue gases per hour (volume at standard conditions) = 769.5 cu ft [item (6)]. Pressure of gases at inlet to exhauster = 3.5 in. water below atmospheric or 29.50 — 0.26 = 29.24 in. mercury, absolute. Saturated vapor pressure at 97 F = 1.763 in. mercury. Pressure of non-condensable gases = 29.24 — 1.76 = 27.48 in. mercury. Volume of saturated gases entering exhauster = $\dfrac{460 + 97}{460 + 60} \times \dfrac{30}{27.48} \times 769.5 = 899.8$ cu ft per hour. Density of saturated vapor at 97 F = 0.002621 lb per cubic foot. Water vapor entering flue = $899.8 \times 0.002621 = 2.36$ lb per hour.

Water vapor whose heat of vaporization and heat of liquid to 72 F is lost = total vapor entering exhauster minus water vapor supplied to system by air and gas minus water vapor which would have escaped if gas had been burned at 100 per cent efficiency = 2.36 — 0.05 — 0.387 — 0.54 = 1.38 lb per hour. Total heat lost as heat of vaporization; $1.38 \times 1038.2 = 1432.7$ Btu per hour. Heat lost as heat of liquid = 1.38 (97 — 72) = 34.5 Btu per hour. Heat lost as sensible heat of non-condensable gases = $880.6 \times 0.06850 \times 0.24 \times (97 - 72) = 361.8$ Btu per hour. Total heat lost through flue = 1432.7 + 34.5 + 361.8 = 1829 Btu per hour. Heat loss in overflow from generator = 36 Btu per hour [item (8)]. Total heat loss through flue if gases had been cooled to 97 F at the exhauster = 1865 Btu per hour.

Results

With an exhaust temperature of 130 F at exhauster outlet the efficiency of the system $= \dfrac{\text{heat supplied} - \text{heat lost}}{\text{heat supplied}} = \dfrac{76{,}842 - 6{,}816}{76{,}842} = 91.1$ per cent based on flue gas analysis.

For checking purposes, the efficiency of the system based on Pitot tube readings and calculations is next determined as follows: $\dfrac{76{,}842 - 7{,}050}{76{,}842} = 90.8$ per cent. [See items (13) to (19) inclusive].

The efficiency of system based on duct method is next found from items (20), (21), (22) and (23). By the duct method the total heat accounted for leaving the system = 70,435 + 7025 + 36 = 77,496 Btu per hour. Efficiency of system = $\dfrac{\text{heat delivered by radiator}}{\text{heat supplied to system}} = \dfrac{67{,}920}{76{,}842} = 88.4$ per cent.

Based on outlet temperature of 97 F at the exhauster, which temperature was attained on actual installations, the efficiency of system based on flue-gas-analysis method [see item (24)] indicates a possible efficiency of 97.6 per cent.

Durability

Past experience with the use of products of combustion associated with water vapors very probably causes one to ask the question, "What about corrosion?" Observations extending over a period of two years would indicate that where the fuel is natural gas, coal or coke oven gas, burned under the conditions of this test, the rate of penetration indicates a long life which is comparable to other types of heating systems. Examination of a generator interior, after almost a year of use with Chicago artificial gas, shows a degree of corrosion that is surprisingly small.

Practically all heating systems rust somewhat inside and usually at a definite rate in inches of penetration per year, depending of course upon the kind of metals and how much the system is used. Because some smokestack breechings have been known to rust in spots under certain circumstances after a comparatively short period of use, one naturally envisions the possible effect of circulating stack gases through pipes.

All stack breechings do not give corrosion trouble nor do all fuels have the same percentage of harmful ingredients in them. Coal as a rule has a very high percentage of sulphur in its products of combustion, which, if combined with moisture and soot, causes mild acids to form and which ordinarily stimulate the rate of rust-

ing above that caused by moisture alone. On the other hand, natural gas as marketed today contains little or no sulphur, and if properly burned produces no soot.

The relative effect of hydrogen-ion concentration on the corrosion of steel and with water which does not contain salts or impurities (which form protective coatings) indicates that at low temperatures the evolution of hydrogen does not increase until a *pH* value of approximately 4 obtains. The *pH* value of the water from various generators which have operated for considerable periods of time with different manufactured gases, showed a *pH* value greater than 4 when measured by the quin-hydrone electrode. The *pH* values obtained from the test set-up were well above 6 on the *pH* scale.

In districts where a poor quality of manufactured gas is produced, it has been found that the hydrogen-ion concentration of the water under the conditions of operation with this generator did not increase enough to increase the rate of action over the neutral water range,

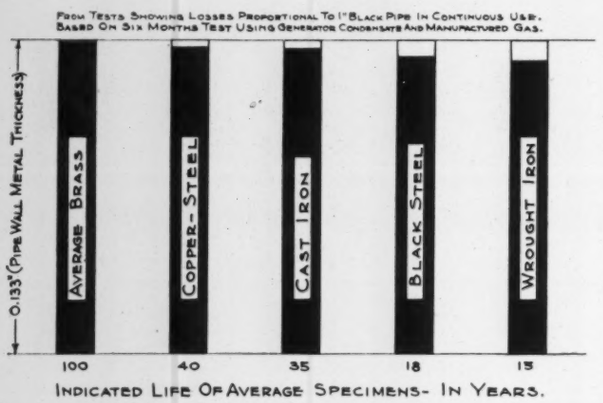


Fig. 7—Penetration of Standard Pipe per Year

and in no case did the water become *bad* from a classifying standpoint due to the acid or scale forming content.

The favorable showing on this corrosion question may be due to several factors, *i. e.*, the constant washing effect from the large volume of vapor, and the continuous flushing through overflow. Also, the fact that the generator operates at a slightly sub-atmospheric pressure with reduced oxygen content (below that of ordinary air) with an increase of inert nitrogen, coupled with the fact that the volume of vapor fills the entire space not occupied by gas, may have a favorable bearing. Again with the perfect combustion which is possible under the controlled air and gas supply, no soot results, and sulphur compounds when present are notably less active at temperatures below 500 F.

As a matter of fact, natural gas is being piped into cities far remote from the source, and better means of purifying manufactured gas is rapidly coming into use, which, with better materials obtainable at low cost, seems to invite serious thought to this more direct method of heating. Tests of metal specimens in solutions of the concentrated generator water for a period of six months are shown in Fig. 7. It will be observed that the steel alloy pipe containing 0.25 per cent copper has more than double the life of wrought iron under these conditions. This is believed to be due to a dense adherent type of rust which prevents scaling and slows up further attack.

There has been no evidence of any appreciable building up of harmful substances over a period of time. Neither has any soot been found or any scaling of pipes where copper steel alloy is used.

Summary and Conclusions

This paper is based on tests made on a specially constructed generator in which gas is burned in the presence of water, thus forming a gas-vapor mixture for heating *radiators*. The most important considerations may be summed up briefly as follows:

1. The elimination of loss in heat transfer through metals at point of heat generation.
2. The reduction in loss of heat from flue gases.
3. Absorption of sensible heat in the gas by conversion to latent heat in vapor which in turn makes it possible to convey heat long distances without appreciable loss in sensible temperature.
4. Reduced temperatures and pressures of gas-vapor, thereby minimizing corrosion as indicated by definite tests showing the rate of penetration per year to be within limits which admit of practical use.
5. Simplicity. An overall efficiency for gas fuel in the 90 per cent range.

It is not the desire to convey the impression that all angles of this method of heating have been investigated. That is conceded as too large an undertaking to complete in two years' time. The first investigations were centered on *results* where it was apparent that additional time was required to prove the *causes*.

Theoretical Gas Efficiencies

Table 1 shows the maximum theoretical efficiencies when burning a typical manufactured gas with various stack temperatures. These are based on the higher heat value of the gas, and do not include any correction for radiation from the boiler covering. Radiation will reduce these efficiencies as much as 5 per cent, depending on the insulating properties of the covering. The gas is assumed to be burned with 35 per cent excess air and has a heat value of 535 Btu per cubic foot at 60 F and 30 in. of mercury. Specific gravity, 0.58; air temperature, 60 F; atmospheric moisture neglected.

Table 1 — Products of Combustion and Theoretical Efficiencies with a Typical Manufactured Gas[a]

	Stack Temperature (F)			
	250	275	300	350
Heat in Dry Flue Gas above 60 F (%)	3.57	4.05	4.52	5.48
Heat in Water Vapor above 60 F (%)	9.13	9.22	9.30	9.47
Heat absorbed by Boiler, or Efficiency (%)	87.30	86.73	86.18	85.05
Radiation (not calculated)	0	0	0	0
Total	100.00	100.00	100.00	100.00

[a] Flue gas analysis: CO_2, 11.18%; O_2, 5.63%; N_2, 83.19%.

It will be seen from Table 1 that the maximum attainable efficiency under practical operating conditions, which include the necessity of having a stack temperature sufficiently high to insure a good draft, and with a normal radiation loss, is approximately 80 per cent.

Report on Code for Testing and Rating Concealed Gravity Type Radiation

COMMITTEE—R. N. TRANE, *Chairman*, E. H. BELING, W. F. GOODNOW, JOHN HOLTON, HUGO HUTZEL, A. P. KRATZ, E. J. VERMERE, O. G. WENDELL

STEAM CODE

A. PURPOSE

1. Object of Code. The object of this code is to provide a practical method for testing and rating concealed radiators which may readily be used by manufacturers and which will provide the necessary information for the heating engineer in the design of the heating plant. This code describes a boiler load method of rating.

2. Definition. The term concealed radiator[1] as used in this code shall apply to any type of radiator installed within a wall, casing or enclosure having an inlet and outlet for the circulation by gravity of the air in the room to be heated.

B. APPARATUS

3. The Purpose of the Test Booth shall be solely to maintain constant and uniform conditions about the radiator.

4. Description of Test Booth.[2] The test booth shall be 12 ft x 15 ft in floor area, and shall have a ceiling height of 9 ft. A tolerance of ± 10 per cent in any dimension is permissible. The floor of the booth shall be set up in a larger room (see par. 5), at least 1 ft above the room floor. One side of the booth (either a 12-ft or a 15-ft side) shall be open with a shield projecting down from the top at least one foot. The walls and ceiling shall have a smooth, close surface, and shall be painted with oil paint. The air in this test booth shall be free from draft.

5. Room Enclosing Test Booth. The test booth shall be set up in a larger room and in such a position as not to be exposed to the direct rays of the sun. The distance between any test booth wall and the wall of the surrounding room shall not be less than 2 ft and the ceiling of the test booth shall not be less than 1 ft from the ceiling of the larger room. The air temperature in this larger room shall be taken at the three sides of the test booth at a level of 5 ft from the floor of the larger booth at a distance of 12 in. from the test booth walls and shall not show a variation to exceed ± 3 deg (F) during the course of a test.

6. Location of Radiator in Booth. The concealed radiator shall be set up in accordance with the manufacturer's specifications, adjacent to the wall opposite the open side of the test booth.

7. Radiator Casing. The radiator shall be placed either in a casing with grilles as furnished by the manufacturer, or if a casing is not furnished, it shall be placed in a box made of material simulating the catalog specifications.

C. GENERAL TEST PROCEDURE

8. Relative Humidity. The relative humidity at the time of test shall be within the limits of 25 per cent and 60 per cent, and the air motion shall be limited to natural circulation.

9. Air Temperature. The heater shall be tested with an inlet air temperature of not less than 60 F nor more than 80 F. These temperatures shall be measured with thermometers placed 12 in. apart (minimum of three) throughout the length of the inlet opening midway between top and bottom, and 3 in. in front of the inlet.

10. Correction for Inlet Air Temperature. When the inlet temperature differs from 65 F, the capacity of the radiator under the test condition shall be converted to the capacity under standard conditions as explained in par. (29).

11. Thermometers. All thermometers used during tests shall be calibrated by comparison with a standard thermometer. They shall be graduated with divisions of not more than 1 deg.

12. No-Load Test. After each test a no-load test shall be run to determine the condensation in the piping alone which must be subtracted from the gross condensation from the radiator in order to obtain the net condensation.

D. PROCEDURE FOR TESTING

13. Piping Diagram. Fig. 1 shows the connections and equipment prescribed for supplying steam to the radiator and measuring the condensate. All instruments shown shall be used and shall be installed in the relative position indicated.

14. The Supply Piping from separator to radiator and return piping from radiator to receiver shall be as short as possible (not exceeding 2 ft in length).

15. The Return Piping from radiator shall pitch downward toward receiver to assure proper draining of condensate. There shall be no intervening valve in the return line between the radiator and receiver.

16. The Receiver. The receiver shall be not greater than 2 in. in diameter and not longer than 14 in. It shall be fitted with a gage glass of outside diameter not exceeding $\frac{5}{8}$ in.

17. Insulation. All piping from the separator to the radiator and from the radiator to and including the receiver shall be insulated with 2 in. of hair felt or equivalent.

18. The Separator shall be of liberal capacity. The steam throttle or low pressure reducing valve shall be of a type suitable for close control. A manometer shall be used on the radiator side of the throttling valve.

19. Steam Temperature. The temperature of the steam shall be measured by a thermometer with the bulb in direct contact with the flow of steam. Thermometers shall be within 6 in. of the radiator inlet and outlet.

20. Steam shall be supplied from a source of sufficient capacity to prevent sudden changes in pressure.

21. Steam Pressure. Radiators shall be tested with steam at 15.7 lb absolute pressure. In order to introduce the required superheat, there shall be not less than 5 lb pressure drop through the throttle valve. If the quality of the steam supplied is so low that the required superheat is not introduced by the 5 lb throttling, the line pressure shall be raised until the required superheat is introduced. The superheat shall be not less than 2 deg nor more than 5 deg and be

[1] The terms *radiation* and *radiator* are used in the commercial sense and not in the scientific sense in these codes.

[2] A refrigerated testing room with walls having a construction typical of walls used in an actual building may be used instead of the warm wall test booth.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

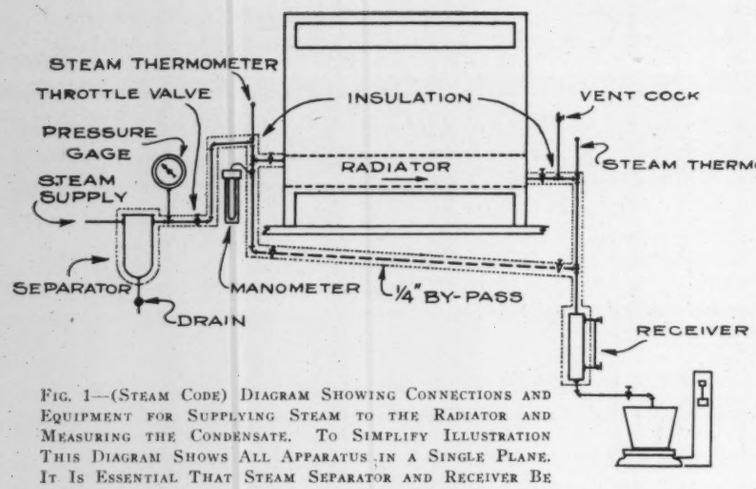


FIG. 1—(STEAM CODE) DIAGRAM SHOWING CONNECTIONS AND EQUIPMENT FOR SUPPLYING STEAM TO THE RADIATOR AND MEASURING THE CONDENSATE. TO SIMPLIFY ILLUSTRATION THIS DIAGRAM SHOWS ALL APPARATUS IN A SINGLE PLANE. IT IS ESSENTIAL THAT STEAM SEPARATOR AND RECEIVER BE LOCATED IMMEDIATELY OUTSIDE OF TEST BOOTH WITH AS SHORT INTERVENING PIPING AS POSSIBLE

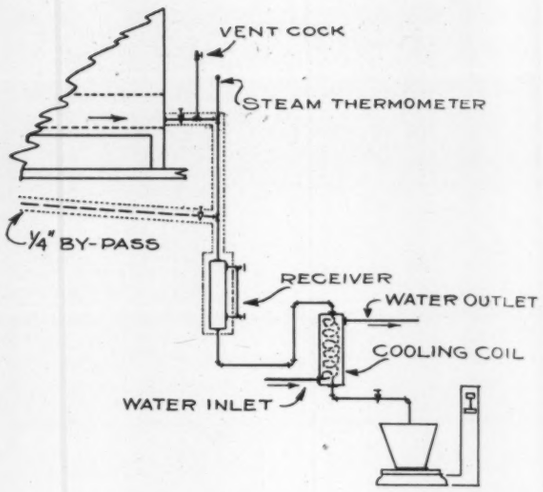


FIG. 2—(STEAM CODE) TEST APPARATUS WITH CONDENSATE COOLER

obtained by an approved method such as wire drawing through a controlled valve from a higher pressure (as shown in Fig. 1).

22. Condensate. The condensate level indicated by the gage glass shall be at the level established at the start of the test at the time of obtaining each periodic condensate weight. At no time during the test shall the water level of the receiver vary more than 2 in. from the level established at the start of the test. The condensate shall pass to the weighing vessel with provision made that no condensate shall be lost by evaporation. The weight of condensate shall be read to 0.01 lb.

23. Air Venting. The radiator shall be vented continuously by means of an air cock through the regular vent opening provided, or through a cock at the top of the condensate receiver. The vent shall discharge outside the radiator enclosure.

24. Start of Test. The test shall begin after the room temperature and condensation rate have remained constant for a period of 30 min.

25. The Separator shall be drained at 5-min. intervals.

26. Duration of Test. A test shall be of not less than one hour duration during which each successive periodic condensate weight shall be within the limits of ±5 per cent of the average of the periodic condensate weights. The condensate shall be weighed each 10 min during a test.

27. Output Under Test Conditions. The heating capacity of a concealed radiator under test conditions shall be determined by the following formula:

$$H_{ts} = W_s h_{fg} \qquad (1)$$

where

H_{ts} = heat emission for steam under test conditions, Btu per hour
W_s = weight of condensate, pounds per hour
h_{fg} = latent heat of evaporation of steam corresponding to absolute pressure in radiator, Btu per pound

28. Correction Factor. The correction factor (C_s) for converting the output at test conditions to the ouput at standard conditions shall be determined by the folowing formula:

$$C_s = \left[\frac{215.3 - 65}{t_s - t_1}\right]^{1.3} = \left[\frac{150.3}{t_s - t_1}\right]^{1.3*} \qquad (2)$$

where

C_s = correction factor
t_s = saturated steam temperature as of test, degrees Fahrenheit
t_1 = average inlet air temperature as of test, degrees Fahrenheit

29. Output Under Standard Conditions. The output under standard conditions (H_s) shall be determined as follows:

$$H_s = C_s H_{ts} \qquad (3)$$

30. Example. The following example will illustrate the use of the equation in pars. (27), (28) and (29) inclusive:

Let

$W_s = 10.03$ lb
$t_1 = 70.0$ F
$t_s = 215.0$ F
$h_{fg} = 970$ Btu per pound (at 215.0 F)

Then

$H_{ts} = 10.03 \times 970. = 9729$ Btu per hour

$$C_s = \left[\frac{150.3}{215.0 - 70}\right]^{1.3} = 1.048$$

$H_s = 9729 \times 1.048 = 10{,}193$ Btu per hour

31. Output Rating. The heat emission or output of concealed radiators shall be stated in Btu per hour but may also be expressed in equivalent direct radiation (E. D. R.) which is equal to $\frac{H_s}{240}$. Ratings for larger sizes must not be made by extrapolating from results on smaller sizes. However, ratings for intermediate sizes may be obtained by interpolation from the results on adjacent sizes. A sufficient number of sizes should be tested to definitely determine the curve expressing the relation between capacity and size.

32. Description of Grilles. The manufacturer shall state the free air area, type and location of grilles upon which the ratings are based. He shall also state the distance from the bottom of the radiator to the top of the outlet grille which shall be designated as the stack height.

33. Sources of Error in Radiator Testing. The major sources of error in addition to accidental mistakes are as follows:

1 Entrained water brought into the radiator with the steam.
2 Too large a receiver.
3 Loss of condensate during the process of collecting and weighing, either by flashing into steam when pressure is released, or by evaporation from hot water surfaces to the air. This loss may be eliminated by means of a condensate cooler. (See Fig. 2.)
4 Incomplete venting of air from radiator.
5 Excessive air currents within the test booth.
6 Wet and/or insufficient insulation.

* Although the exponent 1.3 applies specifically to direct radiators, it also applies with reasonable accuracy to convectors.

TENTATIVE HOT WATER CODE[3]

A. PURPOSE

1. Object of Code. The object of this code is to provide a practical method for testing and rating concealed radiators which may readily be used by manufacturers and which will provide the necessary information for the heating engineer in the design of the heating plant. This code describes a boiler load method of rating.

2. Definition. The term concealed radiator as used in this code shall apply to any type of radiator installed within a wall, casing or enclosure having an inlet and outlet for the circulation by gravity of the air in the room to be heated.

B. APPARATUS

3. The Purpose of the Test Booth shall be solely to maintain constant and uniform conditions about the radiator.

4. Description of Test Booth. The test booth shall be 12 ft × 15 ft in floor area, and shall have a ceiling height of 9 ft. A tolerance of ± 10 per cent in any dimension is permissible. The floor of the booth shall be set up in a larger room (see par. 5), at least 1 ft above the room floor. One side (either a 12-ft or a 15-ft side) shall be open with a shield projecting down from the top at least 1 ft. The walls and ceiling shall have a smooth, close surface, and shall be painted with oil paint. The air in this test booth shall be free from draft.

5. Room Enclosing Test Booth. The test booth shall be set up in a larger room and in such a position as not to be exposed to the direct rays of the sun. The distance between any test booth wall and the wall of the surrounding room, shall not be less than 2 ft. The ceiling of the test booth shall not be less than 1 ft from the ceiling of the larger room. The air temperature in this larger room shall be taken at the three sides of the test booth at a level of 5 ft from the floor at a distance of 12 in. from the test booth walls and shall not show a variation to exceed ± 3 deg (F) during the course of test.

6. Location of Radiator in Booth. The concealed radiator shall be set up in accordance with the manufacturer's specifications, adjacent to the wall opposite the open side of the test booth.

7. Radiator Casing. The radiator shall be placed either in a casing with grilles as furnished by the manufacturer or if casing is not furnished, it shall be placed in a box made of material simulating the catalog specifications.

8. Apparatus. The apparatus to be used is shown in Fig. 1, in which

A is the hot water tank
B is a valve
C is a union
D is the inlet pressure gage
E is the outlet pressure gage
F is the heater
G is the control valve
H is the scale
J is the bucket
K_1 and K_2 are the locations of thermometers

9. Hot Water Tank. A tank of at least 100-gal capacity set up outside of and near to the test booth with the bottom of the tank at least 6 in. above the radiator shall be used to supply water to the radiator. The tank shall be provided with a float valve or other means to maintain a constant water level and a steam coil or other suitable means for heating the water. The tank shall be connected to the radiator with a pipe equal in diameter to the inlet tapping of the radiator.

10. The Valve and Unions (*B* and *C*) are provided to facilitate the change of radiator without emptying tank.

11. Pressure Gages (*D* and *E*) shall be located as near the radiator as possible and shall be marked for an initial zero reading at the beginning of each test by closing the control valve *G* and opening valve *B*, allowing water to come to rest in both glasses for zero markings. The difference between the level in the two glasses will give the friction loss at any time.

12. Thermometers. The thermometers used during tests shall be calibrated by comparison with a standard thermometer. They shall be graduated to read directly to at least 0.2 deg (F).

C. GENERAL TEST PROCEDURE

13. Relative Humidity. The relative humidity at time of test shall be within the limits of 25 per cent and 60 per cent, and the air motion shall be limited to natural circulation.

14. Air Temperature. The heater shall be tested with an inlet air temperature of not less than 60 F or more than 80 F, this temperature to be measured with thermometers placed 12 in. apart (minimum of three) throughout the length of the inlet opening

[3] In order to make the hot water code a practical and usable instrument, it prescribes that for the purpose of comparison all concealed radiation shall be tested and rated under two sets of standard conditions. The ratings so determined shall be the standard catalogue ratings. The reason for not having prescribed ratings for various entering water temperatures and water temperature drops is that there are so many variables involved that an unreasonable number of tests would be required.

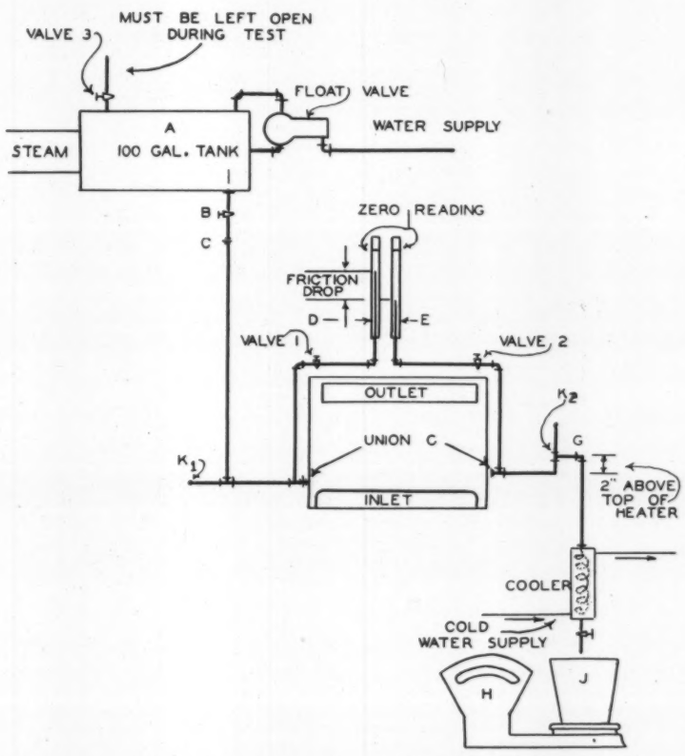


FIG. 1A—APPARATUS FOR HOT WATER TEST

midway between the top and bottom, and 3 in. in front of the inlet.

15. Correction for Inlet Air Temperature. When the inlet temperature differs from 65 F, the capacity of the radiator under test conditions shall be converted to the capacity under standard conditions as explained in par. (25).

D. PROCEDURE FOR TESTING

16. Water Temperature. The temperature of the water shall be measured by thermometers K_1 and K_2 as shown in Fig. 1A with bulbs immersed in direct contact with the water. These thermometers shall be located as near the radiator as possible (within 6 in.).

17. Air Venting. The radiator shall be vented before starting the test by means of an air cock through the regular opening provided.

18. Inlet Water Temperature. The radiators shall be tested with two different inlet water temperatures, namely from 180 to 190 F, and from 130 to 140 F. The maximum allowable variation in the inlet water temperature during the test shall not exceed $\pm$ 1 deg (F) for the test average.

19. Water Temperature Drop. The radiator shall be tested with a water temperature drop of 20 deg (F) with an allowable variation not to exceed $\pm$ 1 deg (F).

20. Start of Test. The temperature of the water in the tank shall be brought to the desired testing temperature (par. 18) and a quantity of water allowed to pass through valve G until the required temperatures are obtained as indicated by thermometers K_1 and K_2. Also the temperature of the test room shall be as indicated in par. (13).

21. Readings. Readings of water temperatures, inlet air temperatures, friction pressure drop and water rate shall be taken simultaneously (within the maximum allowable time of 30 sec.).

22. Duration. The test shall continue until ten sets of readings within prescribed limits as indicated in pars. (18) and (19) have been obtained.

23. Output Under Test Conditions. The capacity of the radiator under the test conditions shall be determined by the following formula:

$$H_{wt} = W_w \ (t_{w_1} - t_{w_2}) \qquad (1)$$

where

H_{wt} = heat emission for hot water under test conditions, Btu per hour

W_w = weight of water passing through radiator, pounds per hour

t_{w_1} = temperature of water entering radiator under test conditions, degrees Fahrenheit

t_{w_2} = temperature of water leaving radiator under test conditions, degrees Fahrenheit

24. Correction Factor. The correction factor (C_w) shall be determined by the following formulae:

For 180 F Inlet

$$C_w = \left[\frac{170 - 65}{\frac{t_{w_1} + t_{w_2}}{2} - t_1} \right]^{1.3} \qquad (2)$$

For 130 F Inlet

$$C_w = \left[\frac{120 - 65}{\frac{t_{w_1} + t_{w_2}}{2} - t_1} \right]^{1.3} \qquad (3)$$

where

t_1 = inlet air temperature, degrees Fahrenheit

25. Output Under Standard Conditions. The output under standard conditions (H_w) shall be determined as follows:

$$H_w = C_w \ H_{wt}$$

26. Example. The following example will illustrate the use of the equations in pars. (23), (24) and (25).

Let

$t_{w_1} = 188$ F
$t_{w_2} = 168$ F
$t_1 = 68$ F
$W_w = 105$ lb

Then

$$H_{wt} = 105 \ (188 - 168) = 2100 \text{ Btu per hour}$$

$$C_w = \left[\frac{170 - 65}{\frac{188 - 168}{2} - 68} \right]^{1.3} = 0.944$$

$$H_w = 0.944 \times 2100 = 1982 \text{ Btu per hour}$$

27. The Standard Hot Water Ratings of a radiator shall be expressed in Btu per hour for average water temperatures in the radiator of 170 F and 120 F and an inlet air temperature of 65 F together with the friction pressure drop through the radiator for these standard conditions expressed in inches of water. Ratings at other temperatures and other temperature drops shall be supplied by the manufacturer on request.

28. Description of Grilles. The manufacturer shall state the free air area, type and location of grilles upon which the ratings are based. He shall also state the distance from the bottom of the radiator to the top of the outlet grille which shall be designated as the stack height.

Discussion Hot Water Section

WARREN EWALD: I should like to ask the purpose of the cooler. Isn't that a cooling coil there in which the water from the radiator is cooled before weighing? I do not offhand see the purpose of it.

R. N. TRANE: That is just for high temperature tests. You can use it or not as you need.

PRESIDENT CARRIER: For vaporization and loss of water, is it not?

MR. EWALD: If it is simply to prevent vaporization, that is all right. Of course, it will not change with temperature.

J. F. MCINTIRE: Personally, I believe that this code for hot water testing, as I expressed it at the 1931 annual meeting, is very incomplete. Any system of testing that requires such volumes of water is subject to error that may be avoided by a closed system. The testing I have had an opportunity to observe, wherein an indirect heater was used with a closed system and the temperatures of the inlet and outlet water measured from which the amount of water circulating through the radiator could be calculated, seems to me to be much more accurate. It does not require thousands and thousands of tests to obtain performance curves on radiators; with such a closed system the points fall so consistently on smooth curves that it is possible to get the extremes and a few intermediate points from which very accurate curves can be drawn.

I do not believe the engineer is going to be satisfied with this tentative hot water code. With more general use of forced hot water I do not believe the tempera-

tures are high enough and since it is possible with a slower system to get such accurate results, I would dislike to see us adopt a code which limits to this method of testing. I feel that we should require complete performance curves on hot water because with the concealed type of radiation the performance is a very important factor.

F. D. MENSING: I would like first to ask the purpose of this code.

MR. TRANE: The purpose is for rating so as to provide information as to what the performance will be.

MR. MENSING: That is what I want to get on record. I don't think anybody here will get information enough on two points to know what a radiator will do. I will agree with all that has been said by the previous speaker. What we need in hot water is not what the radiator will do from 212 F down to two points, but at all points. Anybody who has had experience with radiators of this type alongside of radiators of the other types knows the trouble that the engineer, the contractor and everybody else faces. In the first place, the condition is one which will never be encountered in practice. In a gravity hot water system each radiator is its own circuit. In this case, it is a forced system and the forced system data are given for two temperatures and the chances are that temperatures higher than the highest temperature given would be involved. On a gravity system it does not go low enough nor give the intermediate points and it would only be confusing, not at all helpful, and positively dangerous as far as I can see. There are today in general distribution by radiator manufacturers data given on radiators for hot water, and the same methods applied to this type of radiator would supply all the necessary information. Therefore, I personally do not feel we should accept this section of this code.

E. G. SMITH: Mr. Mensing thought the code did not go low enough for gravity circulation. Of course, in mild weather the gravity temperature is lower and also in a forced system that is true, but it seems to me that the temperature differential does not go high enough rather than that the average temperature does not go low enough. That is, if a gravity system is well designed some of the temperature differentials will certainly be above 30 deg; in other words, those close to the boiler and those farther away will be lower, and it seemed to me it might be well to take them into account, although I do not know that it would make any great difference.

Also, I do not know whether this is of great importance or not either, but there might be an appreciable difference in the efficiency of the radiator, depending on how it is connected. There are three possible ways: *first,* both connections at the bottom, as shown in the sketch here; *second,* one at the top and one at the bottom, but on opposite ends; and *third,* one at the top and one at the bottom, but on the same end. It would seem to me that it might be a good plan to allow any of these connections to be made provided it was specified which was used in the test.

MR. MCINTIRE: I would like to take exception to one more item and that is the assumption of a drop of 20 deg across a radiator, which is not fair to many types of radiators because you do not get that drop. I believe that we should assume a mean temperature of the radiator rather than a fixed drop across the radiator and I think that Professor Giesecke, who has done so much work on pipe sizes for hot water, should express himself as to whether or not he feels that the 20-deg drop across a radiator is desirable or whether it is better to express it in terms of mean temperature.

F. E. GIESECKE: I think the temperature drop should be variable and there should be a much smaller drop than 20 deg with forced circulation. As I understood the committee, their argument was simply to propose a code so that all manufacturers tested under the same conditions so as to get comparative results from different manufacturers. The information for the engineer is probably not as good as might be obtained with a different arrangement, but as I understand the committee, they had in mind a code for testing which would give comparative results on different radiators.

MR. MCINTIRE: In answer to that I may say that it would not be desirable to design on the basis of a 20-deg drop for both a very short and a long radiator. I do not believe that the information that would be secured for a range of radiator sizes would be what you would want. You would want the mean temperature, would you not?

PROFESSOR GIESECKE: I always work on the mean temperature and the rate of circulation of water against temperature drop and, of course, that rate is there, for instance, the temperature drop with forced circulation would vary with the mean temperature, because the heat demanded would vary. For that reason it seems undesirable to provide for a 20-deg drop both at the high and at the low temperature.

H. M. NOBIS: The method of testing radiators could be simplified. I believe that it can be done electrically, thereby giving the complete heat input; then it would be possible to check very easily for both steam and water with the same apparatus. It is very simply constructed. I can give the committee a sketch if they want it.

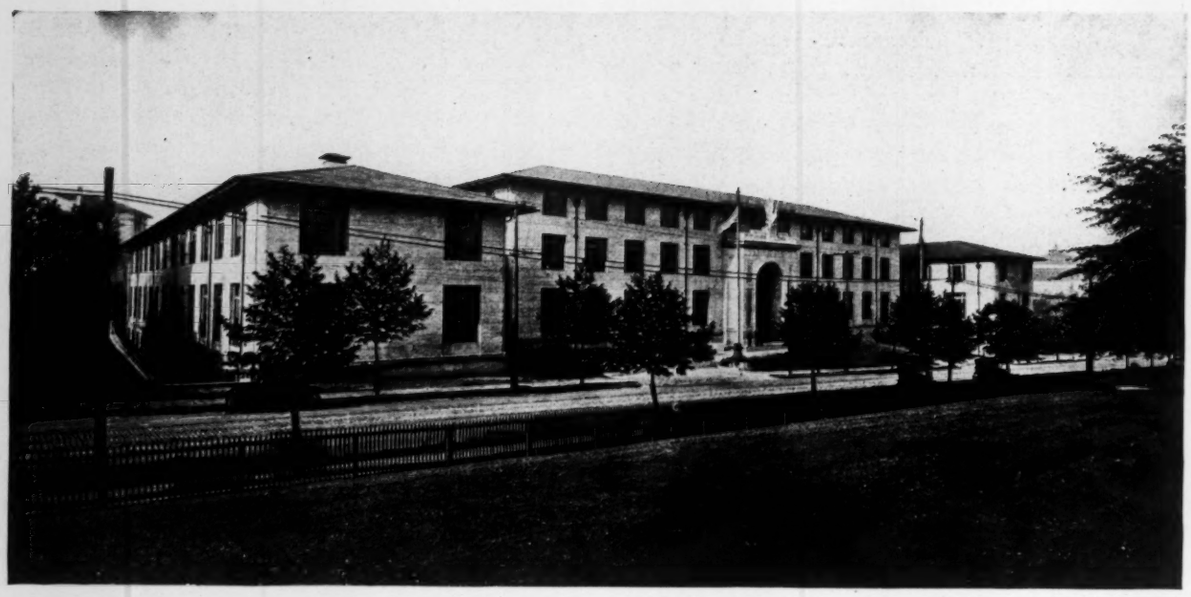
PROFESSOR GIESECKE: I wrote Mr. Trane the same thing some time ago. It seemed to me that by using electricity for supplying energy to the system and having a connection so as to get the no-load effect, as was suggested a while ago, and making a preliminary run, the result would certainly be much more accurate.

M. G. STEELE: I have had considerable experience with precisely that method of testing hot water radiators. It enables one to make a large number of tests at the different temperatures easily and quickly and it seems to be highly accurate.

PRESIDENT CARRIER: This is quite a radical departure from the code as proposed and seems to have a good deal of authority back of it. Do you wish this matter put to vote or do you wish to consider the matter further?

MR. TRANE: I think the questions indicate the complexity of the work of the committee and indicate that we need more talent on the committee. It is a very complicated problem and we never thought for a minute that it would go through as it is. We need to work on it a great deal more.

PRESIDENT CARRIER: Perhaps a little more research? If that is your wish, then we will not put this to vote at this time, but leave it for further consideration at the next meeting.



PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Report of the A. S. H. V. E. Research Laboratory

(June 1, 1931)

By F. C. Houghten, *Director*

SINCE the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, progress has been continued in connection with the twelve research projects which the Committee on Research has under consideration. Active work has been carried on in the Laboratory at Pittsburgh or in the co-operating university laboratories on most of the projects. Those not actively under investigation in the Laboratory have been given consideration by the Technical Advisory Committees with a view to developing plans for the future.

The work has resulted in the publication, or release, of ten technical papers as follows:

1. Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant. (Published in the *American Journal of Hygiene;* Vol. 13, March 1931).
2. Sizing Pipes and Orifices for Gravity Hot Water Heating Systems,* by Elmer G. Smith.
3. Friction Heads in One Inch Standard Cast-Iron Tees,* by F. E. Giesecke and W. H. Badgett [See JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), May 1931.]
4. Air Infiltration through Double Hung Wood Windows,* by G. L. Larson, D. W. Nelson and R. W. Kubasta.
5. Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley.
6. Measurement of the Flow of Air through Registers and Grilles (*Part II*), by L. E. Davies. [See JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), April 1931.]
7. Steam Condensation an Inverse Index of Heating Effect,* by A. P. Kratz and M. K. Fahnestock.
8. Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers,* by L. E. Seeley and E. J. Tavanlar. [See JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), May 1931.]
9. Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems,* by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. [See JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), June 1931.]
10. Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet.

*Papers presented at Semi-Annual Meeting, 1931.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

1. HEAT TRANSMISSION

(Heat Received by and Emitted by Buildings in Relation to Living Comfort)

TECHNICAL ADVISORY COMMITTEE

L. A. Harding, *Chairman*
R. E. Backstrom
F. B. Rowley
A. E. Stacey
J. H. Walker

a. Determination of heat transfer through walls by the Guarded Hot Box Method.

This work was continued at the University of Minnesota under the direction of F. B. Rowley, in co-operation with the

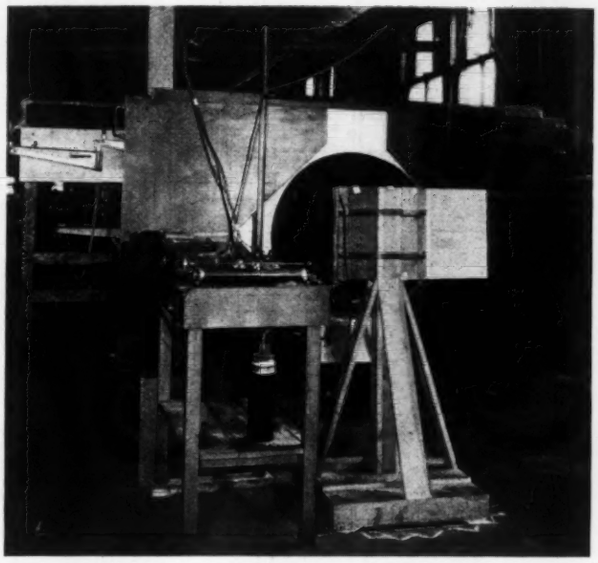


FIG. 1—SET-UP USED IN THE STUDY OF THE EFFECT OF ANGLE OF IMPINGEMENT ON FILM CONDUCTANCE COEFFICIENTS. THE SURFACE BEING STUDIED IS PLACED AT THE OUTLET OF A 30-IN. DUCT THROUGH WHICH AIR IS BLOWN AT A CONSTANT VELOCITY. THE ANGLE BETWEEN THE PLANE SURFACE STUDIED AND THE WIND DIRECTION CAN BE MADE ANY DESIRED SIZE

Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. A number of concrete and stone masonry walls were tested, giving additional data for publication.

b. Film conductance coefficients for various types of surfaces with still and moving air.

This work was continued at the University of Minnesota in co-operation with the Laboratory. Since the publication of the paper Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw [see JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), June 1930], additional data have been collected giving the effect of angles of wind impingement on the film conductance coefficient. A paper on this phase of the work was recently released for publication. Fig. 1 is the set-up used for this work.

c. Effect of Ageing on Conductivity of Concrete.

Additional tests were made in the Laboratory at Pittsburgh to determine the variation in conductivity of concrete with time. Fig. 2 is the set-up used, and Fig. 3 shows the variation in conductivity of the two slabs with the time elapsed since their pouring in the fall of 1927. A paper dealing with this variation, and discussing other factors affecting the conductivity of concrete, has been released for publication. This paper gives a review of conductivity values published by various investigators.

d. Effect of Heat Capacity of a Structure on Heat Flow into a Building from Solar Radiation.

Since the annual meeting, the Laboratory has worked on the analysis of the available data on this subject. A comprehensive plan for additional work to be carried on during the summer months has been outlined by the Technical Advisory Committee.

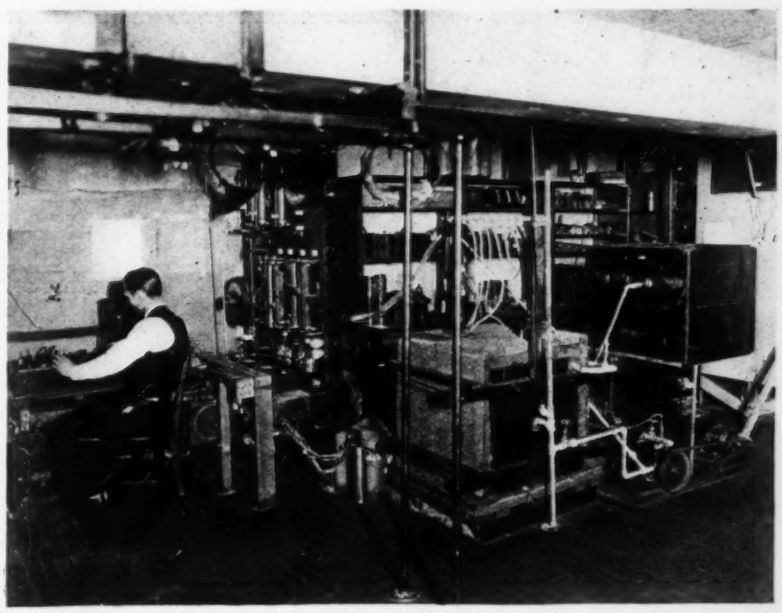


FIG. 2—SET-UP USED IN THE LABORATORY IN PITTSBURGH FOR DETERMINING THE CONDUCTIVITY OF CONCRETE AND ITS CHANGE WITH TIME

2. VENTILATION OF GARAGE AND BUS TERMINALS

Technical Advisory Committee

E. K. Campbell, *Chairman*
D. S. Boyden
H. P. Gant
W. T. Jones
W. C. Randall

A study of garage ventilation was undertaken last year at Washington University, St. Louis, in co-operation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The object of the study was to develop the relations between the quantity of air handled by the ventilating system, its temperature and velocity, the location of intakes and exhausts, the quantity of outside air handled and the concentration and distribution of carbon monoxide and gasoline vapors in the garage. An effort is being made to develop a co-operative study of the subject with some institution located in or near a large city in a northern climate. It is hoped that plans for this work will be perfected in time to begin the study early in the next heating season.

3. PIPE AND TUBING (SIZES) CARRYING LOW PRESSURE STEAM OR HOT WATER

Technical Advisory Committee

S. R. Lewis, *Chairman*
J. C. Fitts
F. E. Giesecke
H. M. Hart
C. A. Hill
A. P. Kratz
W. K. Simpson

a. Capacity of Pipe for Various Parts of a Steam Heating System.

The Laboratory recently completed a study which gave the relation between pressure drop and condensation for various rates of handling air in return risers and dry return mains. (See Fig. 4.) These data are now being further analyzed with a view to revising Pipe Size Tables in the A. S. H. V. E. GUIDE so as to conform with the latest findings of the Laboratory. Additional work is being done at the Laboratory in Pittsburgh using copper pipe and fittings to establish flow relationships for this type of pipe and fittings in steam heating systems.

b. Capacity of Pipe for Various Parts of a Hot Water Heating System.

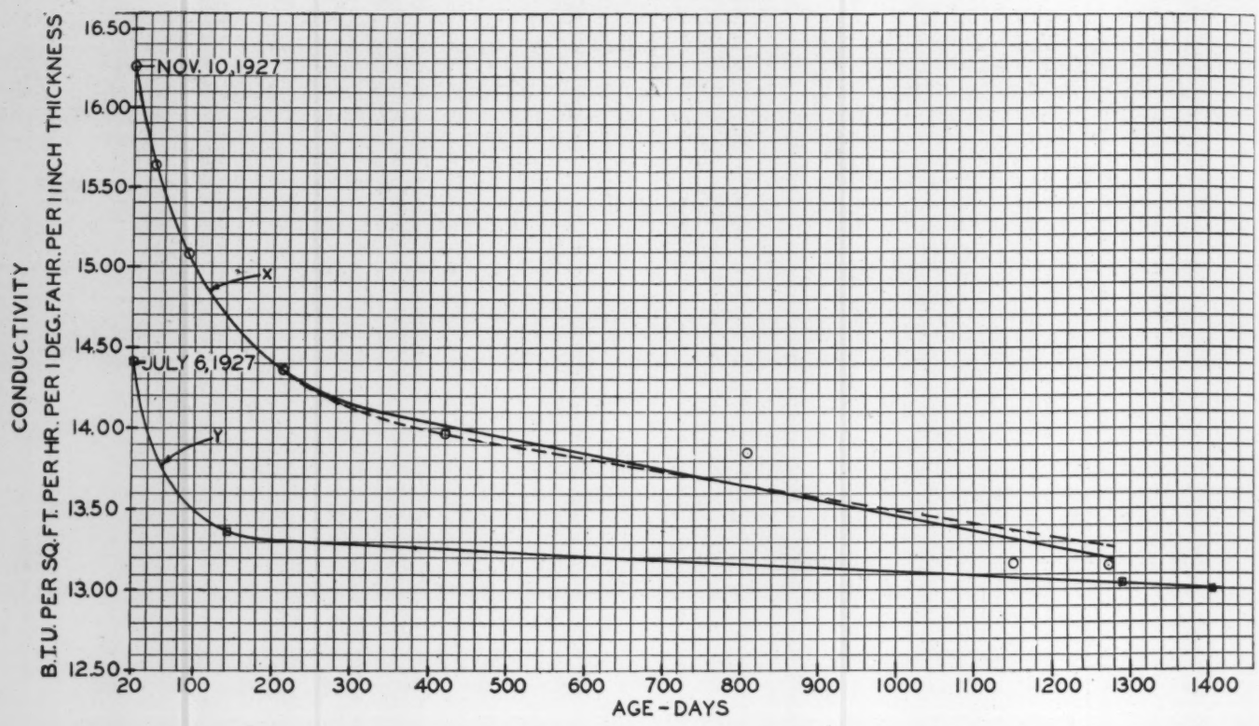


FIG. 3—VARIATION IN CONDUCTIVITY OF CONCRETE WITH AGE, AS DETERMINED IN THE LABORATORY IN PITTSBURGH

This work is being carried on by F. E. Giesecke and his associates at the Agricultural and Mechanical College of Texas in co-operation with the Laboratory. The object of the study is to develop usable pipe size tables for the proper design of all parts of hot water heating systems. See Fig. 5. A paper, Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by Elmer G. Smith, gives usable tables for laying out hot water heating systems and orifice sizes for distributing the flow of water to radiators of various sizes in different parts of the system.

A study is also being made at the Agricultural and Mechanical College of Texas of the frictional resistance to flow of water through the various types of iron, brass and copper fittings on the market. A paper, Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett, gives the frictional resistance for divided flow through cast-iron tees. This work is being continued on other types of tees, including copper, and should give results which have long been needed.

c. Heat Emission from Bare and Covered Copper, Brass and Iron Pipe.

With the increased use of copper and brass pipe in heating systems, there is a growing need for data concerning heat loss from such pipe either bare or covered. Since the external surface area of the copper tubing used is smaller than the same size iron pipe, and since it has a lower emissivity, the heat loss will be less for uncovered copper than for uncovered iron pipe. Determination of these losses is being made in the Laboratory at Pittsburgh.

4. ATMOSPHERIC DUST AND AIR CLEANING DEVICES

Technical Advisory Committee

H. C. Murphy, *Chairman*
Albert Buenger
Philip Drinker
Dr. E. V. Hill
H. B. Meller
Dr. S. W. Wynne

Dean A. S. Langsdorf at Washington University, in co-operation with the Research Laboratory, is undertaking the development of a more sensitive and accurate method of determining air dustiness. This work has been under way during the past year and gives promise of very valuable developments.

The Technical Advisory Committee on this subject has been actively engaged in developing a program for future activity. Its plans include

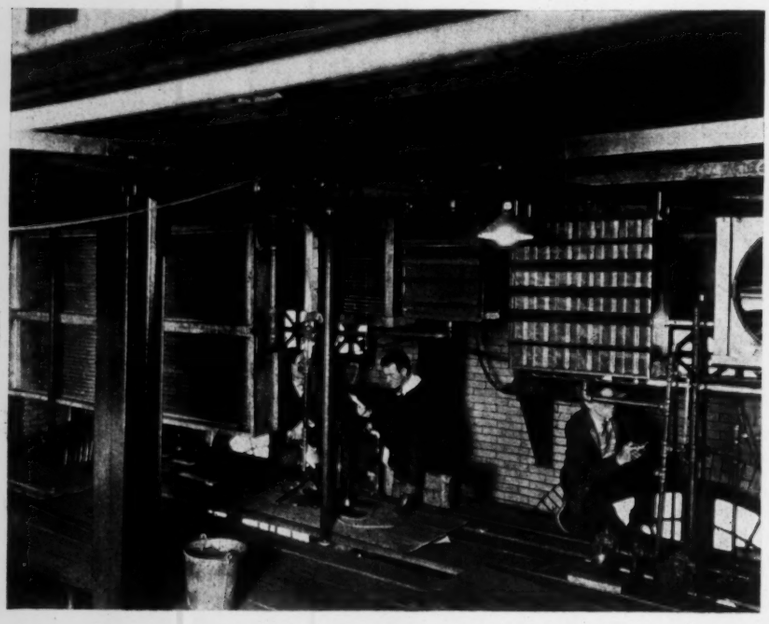


FIG. 4—SET-UP FOR STUDYING PIPE SIZES FOR STEAM HEATING SYSTEMS BY THE RESEARCH LABORATORY IN COOPERATION WITH CARNEGIE INSTITUTE OF TECHNOLOGY, PITTSBURGH. THE UNIT HEATERS ARE USED FOR CONDENSING STEAM FOR STUDYING THE CAPACITY OF THE RETURN SIDE OF A SYSTEM

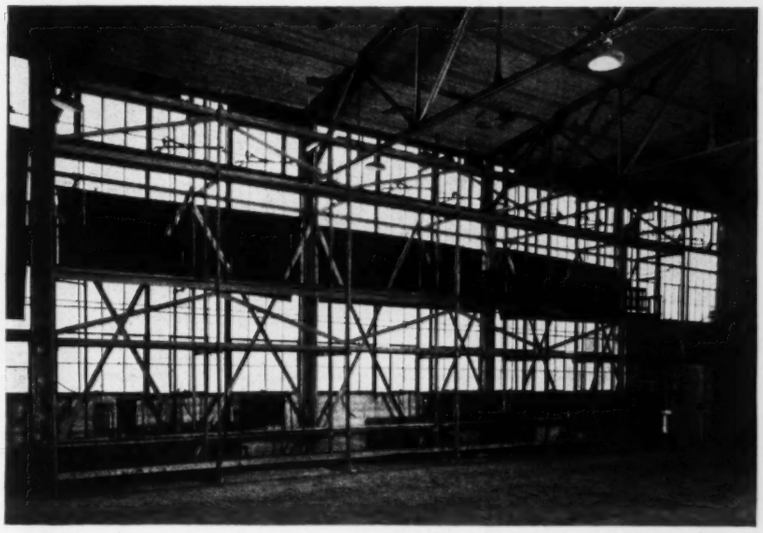


FIG. 5—SET-UP AT AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS FOR A STUDY OF PIPE SIZES IN COOPERATION WITH THE RESEARCH LABORATORY. THIS SET-UP WAS USED IN THE DEVELOPMENT OF DATA ON WHICH TO BASE TABLES FOR DESIGNING HOT WATER HEATING SYSTEMS

the development of a code for testing air-cleaning devices, but since no instrument now on the market is entirely satisfactory for all purposes, it is proposed by the committee to cover by specifications the design of an instrument of a jet-impingement type for industrial and field work, and of an instrument of the filter type for investigation of atmospheric pollution. The committee hopes to correlate the various interests and activities in the field of atmospheric dust and smoke study.

5. RADIATION

(Direct and Indirect)

Technical Advisory Committee

John Holton, *Chairman*
R. M. Conner
R. E. Daley
R. V. Frost
F. E. Giesecke
H. F. Hutzel
A. P. Kratz
J. F. McIntire
W. T. Miller
R. N. Trane

The Research Laboratory recently entered into a co-operative agreement whereby the Laboratory's research work dealing with the heating effects of direct and indirect radiation will be studied at the University of Illinois. This arrangement makes available the excellent facilities for studying radiation developed at the University of Illinois during the past several years.

A study of the heating effect of radiators will be made in the special test rooms, shown in Figs. 6 and 7. This includes a continuation of the study in the room-heating testing plant of the performance characteristics of the various types of cast-iron radiators and also of various types of non-ferrous radiators having heating elements made of copper, brass, and aluminum assembled in enclosures acting as convectors.

These tests are conducted under heating service conditions in actual rooms with outside air temperatures of approximately zero degrees Fahrenheit and with inside air temperatures taken at the 30-in. level, of approximately 68 F. Comparisons between the different radiators tested are made on the basis of weight of steam condensed per hour and the air temperature gradient between ceiling and floor levels. It should be noted especially that, while the air temperature at the 5-ft or breathing-line level is always recorded, it is the air temperature at the 30-in. level which is kept the same, at 68 F, and is used as the basis for comparison.

Professor Willard is also proceeding with the erection and operation of a test booth in accordance with the specifications contained in the Code for Testing and Rating Concealed Gravity Type Radiation (Edition of June, 1931). In this test booth the same radiators which have been tested in the room heating testing plant, will be again tested in accordance with the proposed code for the purpose of obtaining correlation factors between the two plants with various types of radiators.

6. INFILTRATION IN BUILDINGS

Technical Advisory Committee

G. L. Larson, *Chairman*
J. E. Emswiler
F. E. Giesecke
W. C. Randall
W. A. Rowe
J. G. Shodron
Ernest Szekely
M. S. Wunderlich

The Research Laboratory's program for testing various types of walls for air leakage, as outlined several years ago and studied in the Laboratory at Pittsburgh and by Prof. G. L. Larson at the University of Wisconsin in co-operation with the Laboratory, was completed last year. The work at Wisconsin has been aimed toward the clearing up of certain phases of air leakage around weather-stripped and non-weather-stripped windows. The paper Air Infiltration through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta, includes the results of this investigation.

Professor Larson is also making a study of the

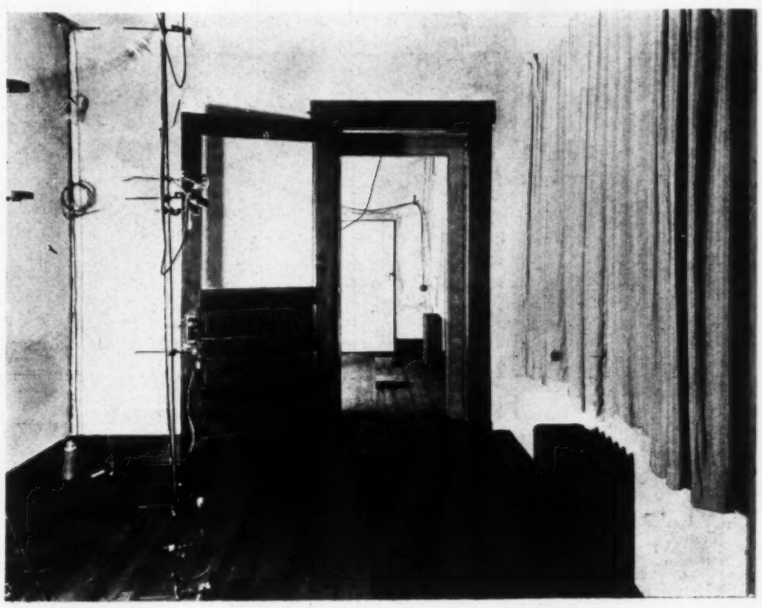


FIG. 6—INSIDE VIEW OF TEST ROOMS IN ROOM HEATING TESTING PLANT, UNIVERSITY OF ILLINOIS. A STUDY OF THE HEATING EFFECT OF RADIATORS IS BEING MADE IN THESE ROOMS BY THE UNIVERSITY OF ILLINOIS IN COOPERATION WITH THE RESEARCH LABORATORY

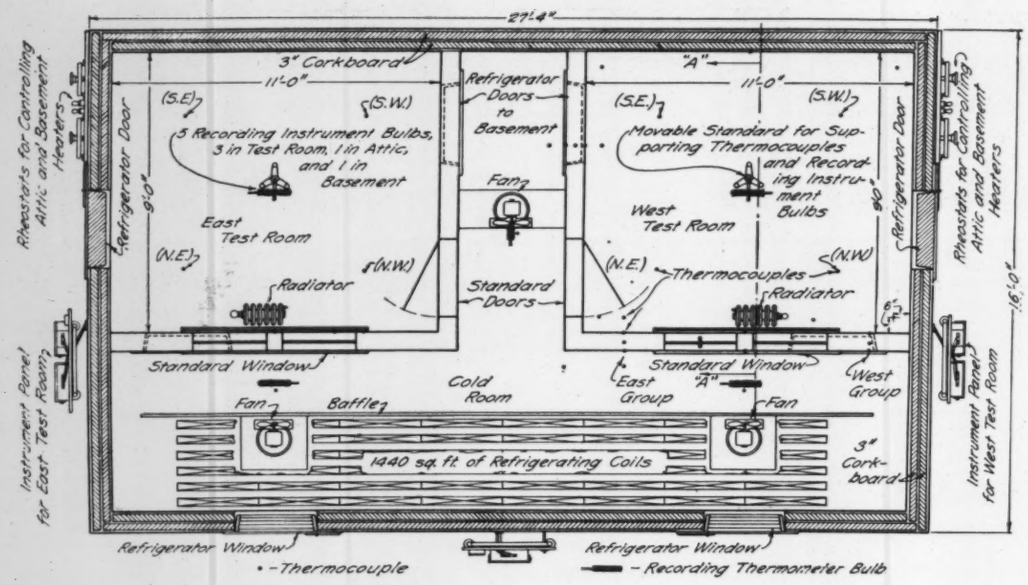


FIG. 7—PLAN OF ROOM HEATING TESTING PLANT, UNIVERSITY OF ILLINOIS. A STUDY OF THE HEATING EFFECT OF RADIATORS IS BEING MADE IN THESE ROOMS BY THE UNIVERSITY IN COOPERATION WITH THE RESEARCH LABORATORY

airation of buildings with special reference to school-type buildings and open window ventilation. Readings are taken to determine the variation in the amount of air entering or leaving through an open window at various wind velocities. For this study, two classrooms of about the same size have been selected, one on the north side and one on the south side of the second floor of a four-story building which has a fully exposed and heated basement having outside openings. It is assumed that because the rooms are near the center of the height of the building the *chimney effect* is as small as it would be at any other place in the building. The inlet ducts from the ventilating system are kept closed; the outlet ducts so far have been kept open. One window in each room is opened a definite height as used for open window ventilation. An anemometer is set in each opening and readings are taken at five minute intervals. At the end of each half hour, anemometers are interchanged so as to eliminate any difference in the instruments.

7. MEASUREMENT OF AIR FLOW THROUGH REGISTERS AND GRILLES

Technical Advisory Committee

John Howatt, *Chairman*
John Aeberly
C. A. Booth
L. E. Davies
D. E. French
J. J. Haines

Prof. L. E. Davies at Armour Institute of Technology, Chicago, in co-opereation with the Research Laboratory is continuing the study of air flow through registers and grilles. Since the 37th Annual Meeting of the Society this study has been directed toward the development of a satisfactory method of measuring the air delivery of a unit ventilator. (See Fig. 8.) In connection with the study of air delivery from unit ventilators and the study of air delivery through grilles into a room, further studies have been made of the application of the formula for measuring air delivery through grilles presented in the paper, Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies. [See JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*) January 1930.]

In most cases the formula has been found to apply satisfactorily, but in the following few cases it does not seem adequate:

1. Where the grille design is characterized by largeness of details
2. Where it is difficult or impossible to determine accurately the amount of free area, as in ornate designs or complex structures such as unit heaters or ventilators;
3. Where the nature of the approach is such as to cause a very erratic velocity distribution across the face of the grille.

A study has been made of the application of a cone, placed over the grille work, into which the anemometer is fitted to measure the air delivery from such grilles as in the foregoing cases. The use of such a cone offers a possibility of success. A paper, The Measurement of the Flow of Air through Registers and Grilles (*Part II*), by L. E. Davies, covers additional phases of the work carried on at Armour Institute.

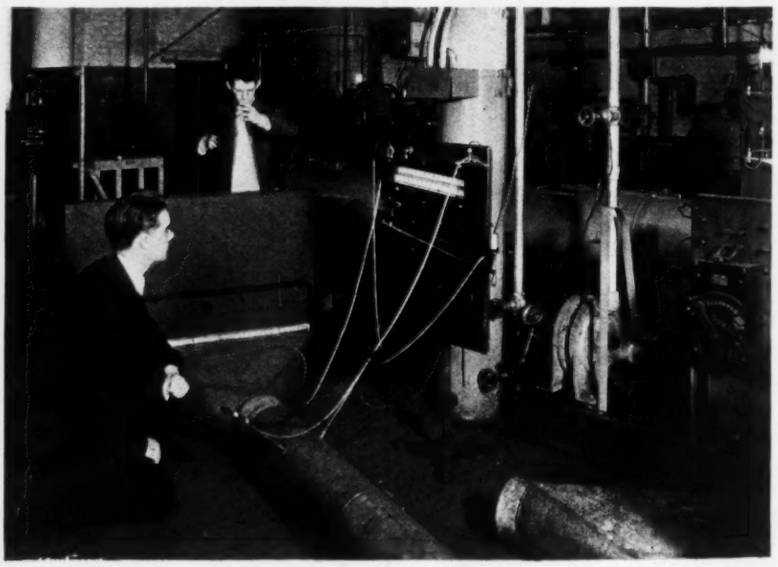


FIG. 8—PROFESSOR DAVIES AND ASSISTANT MAKING DETERMINATIONS OF AIR DELIVERY BY UNIT VENTILATORS, IN COOPERATION WITH THE RESEARCH LABORATORY

8. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT

Technical Advisory Committee

C. P. Yaglou, *Chairman*
O. W. Armspach
W. L. Fleisher
Dr. E. V. Hill
Dr. R. R. Sayers

The work on this study is divided into two distinct phases;

FIG. 9—METABOLIC DETERMINATION MADE IN CONNECTION WITH THE STUDY OF HEAT AND MOISTURE LOSS FROM CHILDREN OF SCHOOL AGE, AND THE APPLICATION OF THE EFFECTIVE TEMPERATURE CHART AND COMFORT ZONE TO CHILDREN OF SCHOOL AGE

first, the study of vital characteristics of the atmosphere by C. P. Yaglou and Philip Drinker at the Harvard School of Public Health, in co-operation with the Research Laboratory; and *second,* a study of the relation between atmospheric conditions and thermal exchanges between the human body and its atmospheric environment made in the Laboratory at Pittsburgh.

The study of atmosphere ionization at the Harvard School of Public Health has resulted in the discovery of a number of very interesting relationships between the ionized condition of the atmosphere and climatic condition, seasonal variation, temperature and sunshine. Other interesting relationships which have been established by the work at Harvard include the effect of methods of ventilation, the washing of air, respiratory and metabolic processes, and room occupancy, on the ionized condition of the atmosphere indoors. A paper, Changes in Ionic Content in Occupied Rooms, by C. P. Yaglou, L. C. Benjamin and S. P. Choate, which deals with these relationships and other interesting facts concerning ionization of atmosphere as a vital characteristic in ventilation, has been prepared for publication. Future plans for this investigation include a more intensive study of physiologic effects of artificially ionized and de-ionized air. This is in addition to the routine daily observations of atmospheric ionization out of doors and indoors, which have been in progress since May, 1930, and which will be continued for at least another year.

It is also hoped that, in connection with this study, daily observations may be made of the atmospheric ionization indoors and outdoors at two other localities, preferably Pittsburgh and Portland, Oregon, in addition to the observations which will be continued in Boston. These data are to be correlated with incidence of illness and death rates in the localities from seasonal diseases in the various localities.

The study of the relation between atmospheric conditions and thermal exchanges between the human body and its atmospheric environment is being continued at Pittsburgh. A paper, Thermal Exchanges between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, resulting from this work, appeared in the March issue of the *American Journal of Hygiene.* Another paper, Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, appeared in the JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), June 1931.

The study of thermal exchanges between the bodies of children of school age (see Figs. 9 and 10) and the atmospheric environment, and the application of the effective temperature scale and comfort zone to children, is now nearly completed, and a paper on this subject will be ready for publication some time during the next few months. Additional plans for this investigation include a study of physiological reactions of persons to low temperatures (within and below the comfort zone) the effect of moisture content of the atmosphere on humans (within the range of 40 to 70 per cent relative humidity); and the effect of cold walls on the comfort zone for persons at rest and normally clothed.

9. OIL AND GAS BURNING DEVICES

Technical Advisory Committee

L. E. Seeley, *Chairman*
R. M. Conner
P. E. Fansler
R. S. Franklin
R. V. Frost
R. C. Morgan
H. F. Tapp

This study is carried on by Prof. L. E. Seeley of Yale University in co-operation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and the *American Oil Burner Association.* The investigation has for its purpose the development of a standard code for testing and possibly rating oil burning boilers and furnaces, and the development of additional facts concerning combustion space, atomization, and air mixture, and their relation to efficient operation.

As a result of this study, a code was presented at the 37th Annual Meeting of the Society for testing low pressure steam heating boilers burning oil fuel. A paper, Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers,* by L. E. Seeley and E. J. Tavanlar, was published in the JOURNAL OF THE SOCIETY (*Heating, Piping and Air Conditioning*), May 1931. The scope of the work in this study is being broadened and will be continued.

* For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

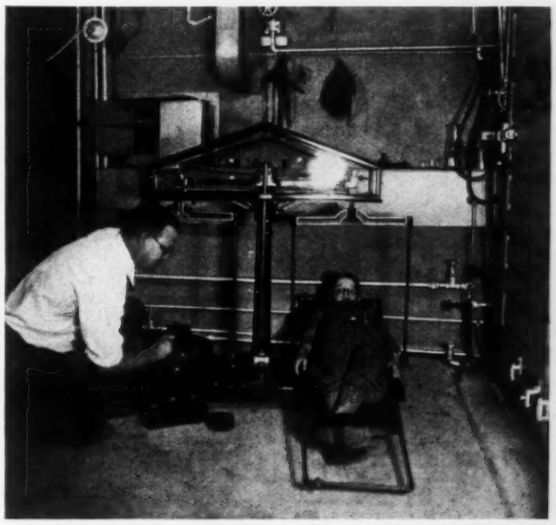


FIG. 10—WEIGHING A CHILD OF SCHOOL AGE IN CONNECTION WITH DETERMINATION OF HEAT AND MOISTURE LOSS FROM CHILDREN OF SCHOOL AGE, AND THE APPLICATION OF THE EFFECTIVE TEMPERATURE SCALE AND COMFORT ZONE TO CHILDREN UNDER SCHOOL CONDITIONS

10. SOUND IN RELATION TO HEATING AND VENTILATION

Technical Advisory Committee

F. B. Rowley, *Chairman*
Carl Ashley
Warren Ewald
R. F. Norris
G. T. Stanton

This is a new project under consideration by the Research Committee, and no definite plan of attack has yet been developed. The study has for its purpose the development of facts concerning noises resulting from heating and ventilating devices and methods of detecting, measuring and locating the source of them.

11. CORRELATING THERMAL RESEARCH

Technical Advisory Committee

R. M. Conner, *Chairman*
D. S. Boyden
P. D. Close
J. C. Fitts
W. T. Jones
H. T. Richardson
Perry West

This is a new committee appointed to receive and administer the work of cataloging and maintaining a file and bureau of information concerning available published data and research activities. This work is now carried on by the *Association for Correlating Thermal Research,* which association will be disorganized after transferring the work to the Research Laboratory and providing for this year's budget.

Program, Semi-Annual Meeting

American Society of Heating and Ventilating Engineers

New Ocean House, Swampscott, Mass., June 22-25, 1931

TECHNICAL

(Daylight Saving Time)

Monday, June 22, 1931

9:30 a. m. Registration.
12:30 p. m. Luncheon (Officers, Council and Authors).
2:00 p. m. Council Meeting.
3:00 p. m. Meeting Committee on Research.

Tuesday, June 23, 1931

8:30 a. m. Registration.
9:30 a. m. Technical Session.
Greeting by D. S. Boyden, Chairman, Committee on Arrangements.
Response by President W. H. Carrier.
Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger.
Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett.
Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by E. G. Smith.
Using Gas-Vapor Mixtures for Heating Purposes, by C. A. Dunham.
Report of Committee on Increase of Membership, C. W. Farrar, Chairman.
Report of Committee on Testing and Rating Unit Ventilators, John Howatt, Chairman.
7:00 p. m. Conference of Chapter Representatives.
7:00 p. m. Meeting of Committee on Ventilation Standards.

Wednesday, June 24, 1931

8:00 a. m. Meeting of Nominating Committee.
9:30 a. m. Technical Session.
Essential Elements for Determining Heating Plant Requirements, by F. B. Rowley.
Report of Committee on Rating and Testing Concealed Gravity Type Radiation, R. N. Trane, Chairman.
Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock.
Experiments to Determine Heating Effect Factor, by R. N. Trane and C. J. Scanlan.
Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar.

Thursday, June 25, 1931

9:30 a. m. Technical Session.
Report of Committee on Ventilation Standards, by W. H. Driscoll, Chairman.
Report of Committee on Research, C. V. Haynes, Chairman.
Reports of Technical Advisory Committees.
Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant.
Infiltration through Double Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta.
Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies.
Predetermining the Airation of Industrial Buildings, by W. C. Randall and E. W. Conover.
Unfinished Business.
New Business.
Adjournment.

ENTERTAINMENT

Monday, June 22, 1931

12:30 p. m. Luncheon (Officers, Council and Authors).
8:30 p. m. Informal Reception and Dance.

Tuesday, June 23, 1931

10:00 a. m. Clock Golf.
1:30 p. m. Coach and Boat Trip to Marblehead and Gloucester, including the famous North Shore Drive.
2:00 p. m. Golf—Research Cup Tournament.
8:00 p. m. Bridge Party for Ladies.

Wednesday, June 24, 1931

9:00 a. m. Motor Trip to Historical Boston and Boston Navy Yard to inspect the frigate *Constitution* or *Old Ironsides.*
2:00 p. m. Golf—Kickers' Handicap.
7:00 p. m. Banquet and Dance at New Ocean House (informal).

Thursday, June 25, 1931

9:00 a. m. Special Golf Tournament—Ladies.
2:00 p. m. Trip to historic Lexington, Concord and Salem.

MINUTE MAN STATUE, CONCORD

Proceedings of the Semi-Annual Meeting 1931

SAILORS' MEMORIAL, GLOUCESTER

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New Ocean House — Swampscott, Mass. — June 22-25

THREE hundred members and guests responded to the call to meet in Swampscott, Mass., at the New Ocean House, June 22 to 25, when one of the finest technical programs ever prepared for a Society summer meeting was presented.

Special action which was taken by the members included acceptance of the Code for Testing and Rating Concealed Gravity Type Radiation (Steam Section); amendment of Regulations Governing the Committee on Research; and approval of additions to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers.

During the three days all technical sessions were held from 9:30 a.m. until 12:30 p.m., and the papers presented covered a wide range of problems important to heating, ventilating and air conditioning engineers. The sessions were well attended and with the brief presentation of papers by the authors, adequate time was given for oral discussions and the many discussions written in advance added interest to the proceedings by bringing out the important points most effectively. It was the opinion of those present that the papers were of unusual interest and the discussions were exceptionally practical in value.

First Session, Tuesday Morning, June 23

The Semi-Annual Meeting 1931, was called to order by Pres. W. H. Carrier at the New Ocean House, Swampscott, Mass., on Tuesday morning, June 23, and D. S. Boyden, General Chairman of Arrangements, for the Massachusetts Chapter, expressed the wish of the local members that a most enjoyable time be had by everyone when they were away from the Technical Sessions. Mr. Boyden said that it was a great privilege to welcome the Society to Massachusetts on this second occasion of acting as host to the members. He recalled the occasion of the 31st Annual Meeting held in Boston in 1925 and mentioned the progress that had been made by the Society as a result of its deliberations at the Boston Meeting.

Mr. Boyden outlined the program of entertainment that had been prepared by the Local Committee and expressed the hope that all in attendance would enjoy the environment of Swampscott and be benefited by the discussions of papers and other technical matters.

In response, President Carrier expressed the appreciation of the Society to the Massachusetts Chapter for the hearty welcome given and the splendid program of entertainment which would add to the enjoyment of the members' visit and make the meeting a success.

President Carrier then explained the rules under which presentation of papers and discussions would be conducted and introduced S. S. Sanford, Detroit, author of the paper entitled, Flow of Steam Through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger (published in June 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 505).

The author gave a short resume of his paper, illustrated with lantern slides and briefly outlined the findings of the tests which he had conducted.

In opening the paper for discussion, President Carrier stated that a model presentation had been made by the author and congratulated him for the clear and concise manner in which he had given his abstract.

Webster Tallmadge presented a prepared discussion and a written comment by J. A. Donnelly was read by P. D. Close, Technical Secretary of the Society, while A. A. Adler gave his prepared discussion. Perry West inquired about the comparative saving to be expected with zone control systems and ordinary individual radiator control methods.

Mr. Sanford in commenting on Mr. Tallmadge's discussion, stated that tests with thin plate orifices gave very consistent results. To Mr. Donnelly's question on radiator noises, the author stated that orifices had been tried in both cast iron and non-ferrous radiators with excellent results. He thanked Dr. Adler for his com-

ments with reference to the Rateau Curve. To Mr. West's question the author outlined results of a recent investigation made by the *National District Heating Association* on the savings obtained with different types of temperature control.

President Carrier called upon Prof. F. E. Giesecke to present the results of some co-operative research work which he had been doing and which he presented in a paper entitled, Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett (published in May 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 430).

A comprehensive written discussion had been prepared by Prof. M. L. Enger of the University of Illinois and it contained a number of illustrations and formulae.

Prof. A. C. Willard pointed out that the resistances or the losses of the fittings are more important than the straight pipe friction and he commended Professor Giesecke for emphasizing the importance of this effect in the flow of fluids.

President Carrier said that in handling air a similar problem is met but is not as difficult to solve as in steam or hot water heating, where the pipes used are only a fraction of the size of air ducts used for ventilating.

The paper entitled, Sizing Pipes and Orifices for Gravity Hot Water Heating Systems by Prof. E. G. Smith (published in July 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 603), and presented by Professor Smith, gave the results of co-operative research, conducted at the Engineering Experiment Station, Texas A. & M. College, on pipe sizes for hot water heating. The object of the paper was to provide a set of tables of pipe sizes and orifices suitable for open or closed systems.

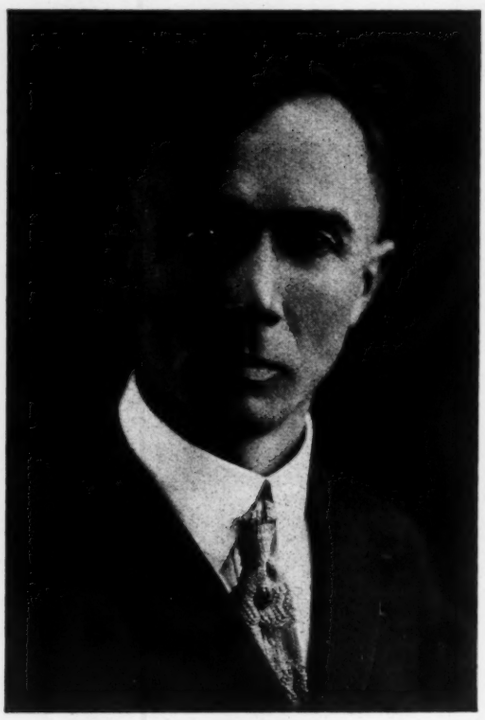
A written discussion was prepared by M. A. Bernhardt, a Philadelphia architect, who felt that the perfection of comparatively simple rules or formulae for accurately designing hot water heating systems would be of immeasurable value to architects and would make the adoption of this system more common for residential work.

Some comments were offered by Dr. A. A. Adler and Perry West stated that the answer to Mr. Bernhardt's questions might be found in the A. S. H. V. E. Guide, although the assistance of an engineer or heating contractor might be required to give the proper interpretation of the hot water heating rules.

R. S. Franklin spoke of his interest in Professor Smith's work and as Chairman of the Standards Committee of the *Heating and Piping Contractors National ssociation*, had adopted the results obtained from this co-operative research as a basis for the preparation of comparatively simple tables for the use of Association members. Mr. Franklin stated that the data had been applied on several trial jobs and the results obtained indicated that a great contribution had been made toward the practical use of hot water circulation for heating.

The final technical paper of the morning was announced by President Carrier who introduced the author for his presentation of the paper entitled, Using Gas-Vapor Mixtures for Heating Purposes, by C. A. Dun-

W. H. Carrier
Newark, N. J.
President



F. B. Rowley
Minneapolis, Minn.
First Vice-President

ham, (published in August, 1931, Journal of the Society, *Heating, Piping and Air Conditioning,* p. 689). Mr. Dunham gave a brief abstract of the paper with some slides showing a new trend in heating results derived from experimental work in which he had been interested. The results of some laboratory tests were given by the author.

Perry West inquired about the corrosion of piping and H. M. Nobis commented on the ingenuity of the method presented and asked if tests had been made with piping systems 100 ft or more long, to see if a separation of gas and vapor occured. F. D. Mensing inquired about sources of the pipe used in the tests.

Mr. Dunham in concluding his remarks described the corrosion studies and stated that the longest pipe run tested was 82 ft.

President Carrier introduced H. A. Snow, Boston, Director of the *Heating and Piping Contractors National Association,* who extended greetings from his Association and best wishes for the success of the Society's meeting, and spoke of the problems of common interest and stressed the effort being made by the Association in establishing closer relationship with other organizations in order to help build up the heating and ventilating industry.

President Carrier expressed the appreciation of the Society for the greetings from the *Heating and Piping Contractors National Association* and assured Mr. Snow of the Society's interest in the conferences and discussions instituted by his Association.

In the absence of John Howatt, Chairman of the Committee on Code for Testing and Rating Unit Ventilators, C. P. Bridges reported that a meeting of the Committee had been held to consider a tentative code but in the absence of unanimous agreement a definite report would not be given to the Society until a later date.

Messages of greeting from Past Presidents R. P. Bolton and J. R. McColl were read to the members after which the meeting adjourned.

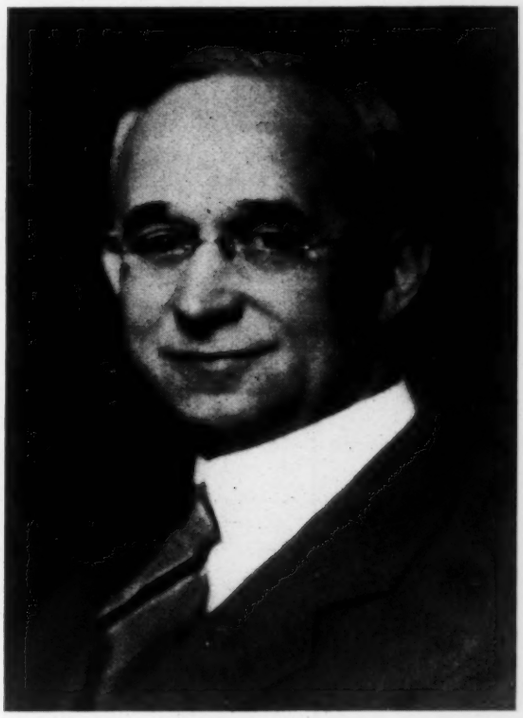
Second Session, Wednesday Morning, June 24

The second session opened on Wednesday morning with the introduction of Prof. F. B. Rowley, who presented his paper entitled, Essential Elements for Determining Heating Plant Requirements by F. B. Rowley, (published in July 1931 Journal of the Society, *Heating, Piping and Air Conditioning,* p. 597). The paper gave a comprehensive outline of the important elements which enter into heating plant requirements and gave a coordinated statement of information taken from various sources for the guidance of engineers in determining heat losses.

In asking for discussion, President Carrier pointed out an important phase that had been covered in this paper relating to the heat capacity of walls. E. K. Campbell related some experiences with tests of intermittently heated buildings particularly those located in southern states.

Vice-President W. T. Jones took the chair and W. L. Fleisher talked on the practical application of the data from the installer's viewpoint, stating that the Society must define good construction and place an average value on it so that the engineer and heating contractor

W. T. Jones
Boston, Mass.
Second Vice-President



C. V. Haynes
Philadelphia, Pa.
Chairman Committee on Research

will be protected against claims for inadequate heating or cooling. W. A. Swain suggested that the assistance of the Weather Bureau be enlisted to obtain standard data on temperatures, humidity, wind velocities and other information needed for determining heating requirements. E. C. Evans thought that co-operation with the architects would produce building designs that would be economical from the heating standpoint.

Chairman Jones announced the report of the Committee on Rating and Testing Concealed Gravity Type Radiation and introduced Chairman R. N. Trane, who presented the Committee's report.

Mr. Trane stated that the instructions given at the 37th Annual Meeting in Pittsburgh had been studied by the Committee and the Code revised so that it was now presented in two sections, a steam code and a tentative hot water code.

Each section being independent of the other, the steam section, he said, would be presented first and the changes from the original draft would be emphasized. In review Mr. Trane said that the Code provided for testing by the boiler load method and proposes rating heaters on the condensation basis. Since the original presentation was approved in principle, he stated, check tests have been made to determine the accuracy of the correction factor 1.3 and Professor Kratz would discuss this matter.

Prof. Kratz, with the aid of slides, presented the results of research on three questions which confronted the Committee after the Annual Meeting, 1931, relating to the construction of the test, both testing in warm and cold walled rooms and the application of the 1.3 power rule. M. G. Steele stated that tests which he had conducted corroborated the statement of Professor Kratz, regarding inlet temperatures of 60 to 90 F. J. F. McIntire inquired about the Committee's reason for having a humidity range of 25 to 60 per cent and thought that the 4 per cent variation shown in tests should be given as 4 per cent plus or minus. After an extended discussion by Messrs. Kratz, Trane and McIntire, L. A. Harding concurred in the belief that manufacturers would test all radiators in accordance with the Code and believed it unnecessary to specify a definite size of unit for test.

A discussion of no-load loss was participated in by Messrs. Steele, McIntire, Kratz and Trane. At the conclusion of the discussion the Committee was requested to take the suggestions offered for the steam heating section of the Code and submit the changes for discussion and approval at the Thursday morning session.

(The complete report of the Committee appears on page 695.)

President Carrier then asked Mr. Trane to present the proposed Code for Hot Water. Numerous questions were asked by Messrs. Ewald, McIntire, Mensing, Professors Giesecke and Smith, Nobis and Steele, with a result that Chairman Trane stated that questions indicate the complex problem before the Committee and further time is needed to solve the very complicated matter. It was therefore, decided to have the Committee do further work on the hot water code and present its report for adoption at a later meeting.

The paper entitled, Steam Condensation an Inverse Index of Heating Effect by A. P. Kratz and M. K. Fahnestock, (published in July 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 621), was presented by Prof. A. P. Kratz, who used slides to illustrate the findings resulting from investigations made at the University of Illinois.

A written discussion by S. R. Lewis offering a new term to express heating effect was presented by P. D. Close at the conclusion of Professor Kratz' presentation. S. S. Sanford inquired whether a change in location of the most satisfactory radiator tested would effect its efficiency. Professor Rowley asked about the relation of the average temperature of the room when the condensation was the same, and R. C. Bolsinger inquired about the relation of radiation and convection factor. Others who participated in the discussion were Elliott Harrington, J. F. McIntire, L. A. Harding and F. D. Mensing.

The paper entitled, Experiments to Determine Heating Effect Factor by R. N. Trane and C. J. Scanlan was given in abstract by Mr. Trane.

Written discussions by R. V. Frost and Profs. A. C. Willard and A. P. Kratz were presented with lantern slides. N. D. Adams read his written comments and the prepared discussions of H. L. Stiles, E. J. Vermere, C. W. Brabbee and P. D. Close were given by Mr. Close.

The final paper of the session entitled, Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers by L. E. Seeley and E. J. Tavanlar, (published in May 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 419), was presented in abstract by Prof. Seeley. This paper, though quickly presented, contained a great deal of practical information of value to engineers interested in boiler and oil burner performance and resulted from extensive investigations at Yale University. The curves showing boiler outputs at which maximum efficiency occured were of particular interest as well as the discussion of sound emitted by the various burners.

Two written discussions were submitted by J. T. Webber and W. F. Clockau. The meeting then adjourned.

Third Session, Thursday Morning, June 25

The final session of the meeting was opened by President Carrier who introduced W. H. Driscoll, Chairman of the Committee on Ventilation Standards, who stated that the Committee was making an informal report of progress and he would outline the work accomplished and the goal of the Committee. The object of the Committee's work he said was to prepare a Code on Ventilating Requirements for the use of legislative bodies and it is intended that the report will set up certain definite standards which legislators, engineers or members of the Society might use as a basis for or to assist in the preparation of bills for enactment that will conform to particular conditions in a given locality. Because of the requirements of different buildings, building construction, climates, etc., a uniform act for adoption of all legislatures would produce endless confusion, Mr. Driscoll pointed out. The standards to be set up, he said, must be sound, reasonable and logical and in order to safeguard the ventilation conditions, particularly in

schools, it is the Committee's hope that its report in final form will be a bulwark against attack. The final report, he stated, will cover all classes of buildings to be ventilated and it is expected that the work of the Committee will result in giving the Society a comprehensive report for discussion at the Annual Meeting next January.

W. A. Rowe suggested the publication of the Committee's report in the Journal in order to get the comment from members who could not attend the meeting. Professor Giesecke pointed out that kitchen and toilet room ventilation sources of contamination should be considered. H. M. Nobis thought school building ventilation should be considered separately to meet the opponents of ventilation, who are concentrating their attacks on schools.

The Chairman explained that the plan was not to set up standards for specific classes of buildings, but to define proper ventilation and permit legislators to determine how and where they will apply standards to various types of buildings.

L. A. Harding thought that the exceptions included in the tentative draft of the Code should be removed and something like the following be used: "This Code shall apply to all space or spaces occupied or intended to be occupied by human beings in any and all buildings for which legislature enactment is to be provided." He also said that the sources of air supply should come from an uncontaminated place but the inclusion of such statements requires definitions in the code, something the Committee wanted to avoid. A constructive suggestion by J. D. Cassell proposed the printing of definitions as an addenda to the Code and was under consideration.

Vice-President F. B. Rowley took the chair, and President Carrier offered some suggestions regarding air distribution, drafts, temperature ranges, control methods and demonstated how bodily heat production and temperature at which the air supply must be introduced make a 25 to 30 cu ft per person per minute air supply a rational standard.

J. W. H. Myrick gave some of the history of Massachusetts' Ventilation Law and related experiences he had as school commissioner with the operation of ventilating. Perry West and W. L. Fleisher offered suggestions respectively on outside air quantities and the significance of carbon dioxide as a measure of air quality.

At the conclusion of the discussion, on motion of Mr. Myrick, properly seconded, it was voted that the Progress Report of the Committee be accepted and sent to all members and that the Committee be asked to carry on its work and render a final report for Society action at the earliest possible date.

President Carrier announced that R. N. Trane would report for the Committee on Code for Testing and Rating Concealed Gravity Type Radiation. Mr. Trane stated that the Committee held a meeting after the Wednesday morning session to discuss the suggestions which had been made and had included them in the report which he presented.

Vice-President F. B. Rowley took the chair, and additional proposals were made by Mr. Fleisher and Professor Smith, regarding the number of tests to be made and by J. F. McIntire regarding the test room to be used.

Professor Rowley then said that as the Code had been adopted in principle, the regular procedure would be followed upon receipt of the Committee's final draft namely: have the Code printed and sent to the membership for approval by letter ballot vote.

Report of Committee on Research

The Chairman of the Committee on Research, C. V. Haynes, was introduced and briefly presented a report of the progress made since the Annual Meeting of the Society. He read a letter which was being sent to manufacturers interested in the Society's Research work and described a booklet being printed for the Committee on Research, giving a list of the projects under way, the history of the Laboratory and an index to papers published in the Society's Transactions and Journals which originated at the Research Laboratory or in co-operative institutions.

Mr. Haynes outlined the problems under investigation by the Committee on Research and announced that a new era in the Society's Research activities was beginning because of the action of the committee in appointing Prof. A. C. Willard as Technical Adviser.

A report prepared by F. C. Houghten, Director of the Research Laboratory, outlining the work at Pittsburgh and in cooperating institutions was presented. (For complete text of report see page 700.)

W. H. Driscoll assumed the chair, and Prof. F. B. Rowley outlined the changes in the regulations governing the Committee on Research which had been approved by the Society in January, 1930. The Committee desired the Society to vote upon a change in *Article 7*, Section 1, specifying that the quorum be seven (7) members. It was requested that an amendment be made so that the quorum for a research meeting be the same as for the Council of the Society, five (5) members. A proposed amendment to *Article 7*, Section 2, suggested the elimination of the following: "All papers, findings or reports resulting from the work of the Research Committee shall when published by the Society be headed by a picture of the Laboratory at Pittsburgh, Pa."

It was also recommended that par. *F* be added to Section 1, making the retiring Chairman of the Committee on Research a member for an additional period of one year even though his three year term of office may have expired. The Committee believed that it is important to have the retiring Chairman remain on the Committee because of his administrative experience.

It was regularly moved and seconded that the changes requested by the Committee on Research be approved by the Society.

Report of Technical Advisory Committee on Radiation

John H. Holton, *Chairman*

Prior to the Summer Meeting, 1931, a letter was sent to all members of this committee outlining the problems to be discussed. Combining replies received from these letters, and personal interviews with members who attended this committee meeting, the following recommendations represent the majority opinion.

RECOMMENDED:

1.—That for the present, no action be taken on the commercial testing of radiation, either exposed or concealed, by

the A. S. H. V. E. Laboratory. Our laboratory facilities can be used to better advantage along lines tending to clarify approved methods of testing all forms of radiation.

2.—That codes for testing all forms of so called "radiation" both exposed and concealed, be developed and approved by the Society. With these as a basis, other agencies may then be in a position to do all the necessary testing, then if desired, the A. S. H. V. E. may consider the proper organization and equipment to undertake this commercial testing. Until such codes are a part of the approved program of the Society, we are not in a position to render needed service to the industry.

3.—That in view of the divided opinion and the many unknown factors involved in the effectiveness of the various heating mediums, it is suggested, at least for the present, we go on record as favoring a condensation method properly controlled as the measurement of the total heat delivery from such equipment. In other words, a condensation rating represents a load method and measures the amount of heat required by, and delivered to the heating equipment.

4.—That the Society continue to authorize the study of the utilization or effectiveness of the heat delivered by various types of heating equipment. This involves further investigations of the so called heating effect, and should determine:

(a) Definition of heating effect
(b) Method of measurement.
(c) Correct location of temperature control point.
(d) Effect of variation in wall construction and wall temperatures. This work must lead to a standard procedure or code for the testing of heating effect as defined.

5.—That in view of the foregoing, a Committee be appointed to revise and bring up to date in the form of a code, the present exposed radiator test method, having in mind the desirability of correlating this code with that now under consideration on concealed radiation. We further recommend that these test codes be published in separate pamphlets available to all members for insertion in their book of standard codes.

6.—That definite tests be authorized to prove whether or not the so called hot room method of testing gives the correct results, bearing in mind that if possible, this method should be approved due to its simplicity and low cost. Such a method would be within reach of all manufacturers, while the high cost of the cold room method would tend to retard its universal acceptance. Such work is, at present, under way in the University of Illinois, and should be continued on both concealed and exposed radiation units.

While this Committee could make further recommendations for additional program at this time, it feels advisable to await the outcome of the development of these codes and test methods before definitely recommending further tests. The results of the above will clearly indicate the trend of our future research work.

Report of Technical Advisory Committee Air Conditions in Relation to Comfort and Health

C. P. Yaglou, *Chairman*

During the past two years the Harvard School of Public Health in cooperation with the Society's Research Laboratory has been studying the problem of ionization in relation to air conditioning and health. Several phases of the work dealing with diurnal and seasonal variations in the ionic content of outdoor and indoor air, ionic changes due to respiratory and metabolic processes, and the de-ionizing effects of air conditioning methods, have already been completed. A paper on these is being prepared for the Annual Meeting, 1932, in January, and it is scheduled for publication in the September issue of the Journal of the Society.

The phases of the work now being studied are:

a. Physiologic, bacteriologic and bio-chemical effects of artificially ionized and de-ionized air.
b. Relationship between seasonal variation in atmospheric ionization and seasonal variation in the incidence of respiratory diseases.
c. Methods for artificial ionization of air.

Other plans of the committee for future work include the physiologic, bacteriologic and bio-chemical effect of cold and of humidity on human beings and animals.

Very little information is available on the effects of cold upon the organism. The region covered by the Research Laboratory extends from 80 deg E. T. upward. In the opinion of the committee, the Research Laboratory should complete this work, which, in conjunction with the study on ionization may throw some light on the etiology of respiratory diseases.

Concerning the influence of humidity on the organism, the committee proposes a thorough study of the effects of changes in humidity through a range of 60 per cent at three different temperatures, namely, 40 deg, 70 deg, and 95 deg F, with the possibility of determining the optimum humidity zone. Observations are to be made on men as well as on animals. The observations on animals are to include the course and resistance to pneumococcus and enteric infections at high and low humidities, growth, reproduction, etc.

Perry West presented the following resolution:

"Whereas, many of the problems concerning the value and worth of ventilation and air conditioning from a comfort standpoint have been solved by our Research Laboratory, and *Whereas,* there remains to be solved the very important problems concerning the value and worth of ventilation and air conditioning from the health, efficiency and longevity basis; *Therefore be it Resolved,* that it is the sense of this meeting that the Society give serious consideration to the establishment of the worth of ventilation and air conditioning from a health and efficiency basis and that the Research Committee be requested to study and report on that matter at the next Annual Meeting."

This resolution was seconded and when voted upon, unanimously approved.

A paper from the Research Laboratory in Pittsburgh was announced by Chairman Rowley and presentation of data on Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, (published in the June 1931 Journal of the Society, *Heating, Piping and Air Conditioning,* p. 493), was made by F. C. Houghten, Director of the Laboratory.

In a brief abstract Mr. Houghten stated that the paper contained additional data on relation of man to his atmospheric environment and presented in practical form a method of applying the effect of heat and moisture loss of the bodies of men working in still air to industrial problems.

A written discussion was presented by W. L. Fleisher.

The paper Infiltration Through Double Hung Wood Windows by G. L. Larson, D. W. Nelson and R. W. Kubasta, (published in the July 1931 Journal of the Society, *Heating, Piping and Air Conditioning,* p. 583), was given in abstract by Mr. Nelson and presented the results of investigations made in co-operation with the Research Laboratory at the University of Wisconsin. W. C. Randall gave a prepared discussion.

A paper entitled, Predetermining the Airation of Industrial Buildings by W. C. Randall and E. W. Conover, (published in the June 1931 Journal of the Society, *Heating, Piping and Air Conditioning,* p. 513), was presented briefly by the author, Mr. Randall, and gave results of investigations being conducted by Prof. Emswiler. Comments were made by L. A. Harding and a written discussion by A. Vogel was presented.

In the absence of Professor Davies, his paper, Measurement of the Flow of Air through Registers and

Grilles by L. E. Davies, (published in the April 1931 Journal of the Society, *Heating, Piping and Air Conditioning*, p. 325), was presented by title and Chairman Rowley announced that prepared discussions had been received from A. Weixel, S. I. Rottmayer, E. H. Beling and E. P. Wells, with replies to Messrs. Beling and Wells by the author.

Under the heading of new business a special committee appointed by the Council to consider the resolution of the *National Association of Fan Manufacturers* with respect to amending the Standard Test Code for Disc and Propeller Fans, reported that it had studied

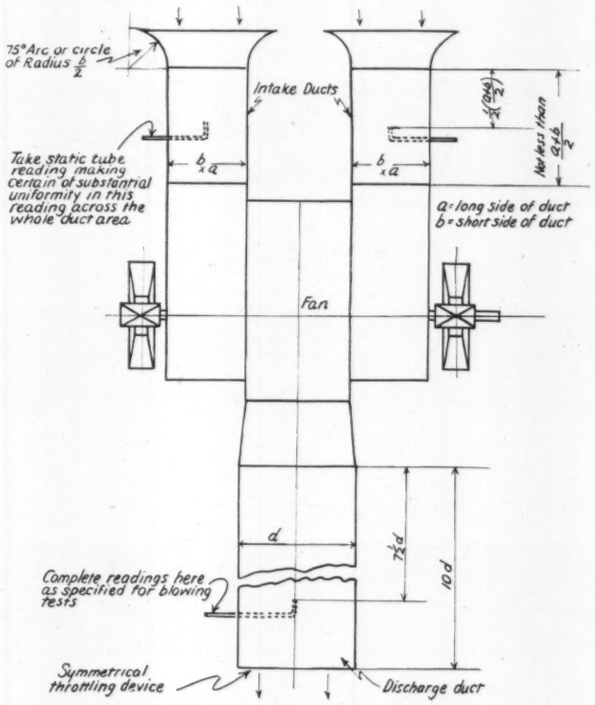

75° Arc or circle of Radius b/2
Intake Ducts
Take static tube reading making certain of substantial uniformity in this reading across the whole duct area
b x a
a = long side of duct
b = short side of duct
Fan
d
7½d
10d
Complete readings here as specified for blowing tests
Symmetrical throttling device
Discharge duct

PLATE *G*

the proposal and on motion of L. A. Harding, seconded by W. T. Jones, it was voted that the following addition be made to the Standard Test Code:

Following the paragraph *Exhausting Fans* insert new paragraph—*Centrifugal Fans with Inlet Boxes:*

Centrifugal fans with inlet boxes shall be tested with inlet boxes in place. A duct (or ducts for double inlet fans) of the same size and form as the inlet box opening shall be connected thereto, as shown on Plate *G*. The inlet test duct shall be not less in length than the mean of its two sides. The open end shall be provided with a nozzle inlet formed of cylindrical sides curved to a radius of not less than one-half of the short dimension of the duct, and not less than a 75 deg arc. Readings shall be taken in said test duct at a point one-half of a mean side from the open end exclusive of the length of the nozzle. The readings are to be taken in accordance with Plate *G* and the calculations made in accordance with the calculation sheet for Centrifugal Fans Exhausting. (See Plates *A, B* and *C*.)

Ratings based on such tests shall be clearly stated as applying to *Centrifugal Fans with Inlet Boxes* and include the loss in the boxes.

Paragraph *Exhausting Fans,* ninth line—place an asterisk after the word *adding* with the following footnote to which it refers:

This addition is algebraic if, instead of calculated absolute pressures, observed readings are used, those below atmospheric pressure being negative, those above atmospheric pressure positive, and the velocity head always positive.

Paragraph *Disc and Propeller Fan Tests* under the sub-heading *Test Setup,* fourth paragraph in last sentence, omit the word *static*. The revised form of this sentence to be—

The pressure in the large chamber shall be determined by the reading of an impact tube. (See Plate *B*.)

Resolutions

Mr. Harding offered the following resolutions which were seconded by H. M. Nobis and unanimously carried:

RESOLVED, That the thanks of the members attending the 37th Semi-Annual Meeting of the Society be officially expressed to the Massachusetts Chapter for the hospitality extended to the Society on this occasion and especially to the local Chapter Committees for the excellent entertainment and program provided; the fine service rendered by the hotel management; the co-operation of the authors of the papers presented and the assistance of the press in giving fine publicity to the proceedings of the meeting and the activities of the Society.

As no further business was offered, President Carrier declared the 37th Semi-Annual Meeting adjourned.

Committees

General Committee for Semi-Annual Meeting 1931
D. S. Boyden, *Chairman*
W. T. Jones, *Vice Chairman*

Reception and Registration
J. S. Webb, *Chairman*
E. V. Cole
J. A. Reardon
W. B. Leland
A. Kellogg
C. R. Swaney

Finance
W. T. Jones, *Chairman*

Ladies and Hostesses
Mrs. E. A. Dusossoit, *Chairman*
Mrs. Leslie Clough
Mrs. W. T. Jones
Mrs. J. H. Brinton
Mrs. T. F. McCoy

Sports
E. A. Dusossoit, *Chairman*
F. R. Ellis

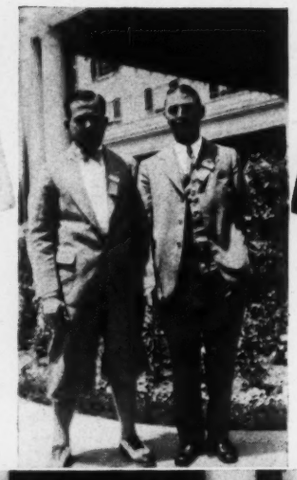
Entertainment
J. H. Brinton, *Chairman*
H. B. Wiegner
W. N. Murray
Coll T. Flint

Publicity
L. Clough, *Chairman*
R. F. Gifford
A. T. Papineau

Golf
J. F. Tuttle, *Chairman*
A. C. Bartlett
R. S. Franklin
A. S. Klonnower
W. R. Illeg

Transportation
T. F. McCoy, *Chairman*
P. C. Pocock
E. W. Berchtold

D. S. Boyden, Boston, Mass.; A. F. Karlson, Fitchburg, Mass.; W. R. Fraser, St. Louis, Mo.; A. W. Moulder, Providence, R. I.

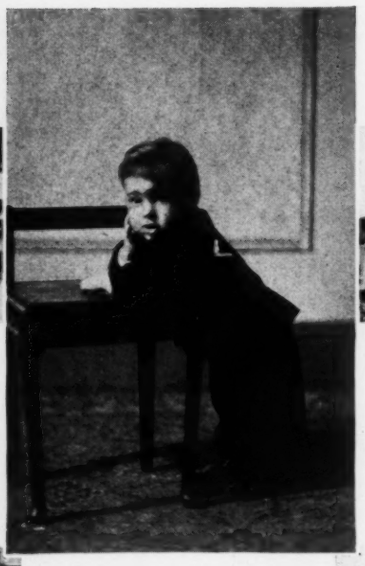


Ralph S. Franklin, Boston, Mass., Winner of the Research Cup. Walter Fleisher, New York; Coll Flint, Boston, Mass.; Stanley Smith, Westfield, Mass.

Above — Merrill Blankin, Philadelphia, Pa.; Warren Webster, Jr., Camden, N. J.

Below—Pres. and Mrs. W. H. Carrier

C. H. Quirk, New York; Wm. H. Price, Jr., Newark, N. J.; J. W. Brinton, Boston, Mass.



P. D. Close, Jr., The Youngest

D. S. Boyden, Chairman General Arrangements; T. F. McCoy, Chairman Transportation; J. S. Webb, Chairman Reception and Registration

R. W. Cumming, New York; C. A. Sawade, Johnston, Pa.; H. W. Hochuli, New York; E. J. Ritchie, New York

Fred Ellis, Boston, Mass.; T. Chester, Detroit, Mich.; L. C. Soule, Newark, N. J.; W. A. Rowe, Detroit, Mich.

. . .V. E. Enjoyed Swampscott

Dear A. S. H.—:

SWEET Spirits of Swampscott! If you don't think that the 1931 Semi-Annual Meeting was a pretty potent party just listen to this tale of an "old salt," who recently mastered his first Ipswich clam with no damage to necktie or vest.

You birds who "can't get away???" to attend a Society Meeting wouldn't know an Ipswich clam from an Abalone, because all of these things are found out at Summer Meetings and lately you have been neglecting your education.

Now, lend me your good ear and I'll whisper in the high voice that the best little spot on Massachusetts Bay is the New Ocean House, where the fireworks commenced on Monday, June 22. The first mess call was a revelation because this place is run on the principle of let eating be unrestrained. We had been hearing about the groceries but wouldn't believe it. Gained a pound a day and now am taking reducing exercises with music every morning at seven.

Dave Boyden had everything organized for the Massachusetts Chapter and his aides were on the job every minute —Colonel's orders. Bill Jones, the skipper from Waban, was adjutant, finance officer and toastmaster at the banquet, a versatile fellow.

Jack Webb (Reception and Registration) was the official greeter and he was there to welcome us and see us go. Ed Dusossoit looked after the sports and Mrs. D. arranged all of the ladies parties while Leslie Clough told the world about what went on at Swampscott through the Boston, Lynn and New York papers.

The celebrities started to arrive early to enjoy the salt air, sunshine and swimming. President Carrier and Perry West rolled up in a Packard and Frank Rowley and Mrs. Rowley flew from Minneapolis in their plane. Past Presidents were all about the lawn, library and lounge; the book worms, Harding and Willard, Homer Addams and Bill Driscoll had their wives and families with them. The Council arrived in good order from all points of the compass, the long distance traveling medal being awarded to E. K. Campbell, Kansas City, with lesser awards to Bill Rowe, Ros Farnham, John Cassell, Fred Mensing, R. H. Carpenter and F. C. McIntosh.

At first it appeared that the Bills' (not Elks) in attendance would outnumber all others, but the Harrys came from behind with Black, Heckroth, Jenkins, Issertell and Wheller, to win the gold plated jelly fish awarded at low tide. Andy Edgar kept his 100% record intact by being on duty for all sessions.

The real big time doings commenced at noon Monday when the Authors, Officers and Council attended a big luncheon party to get acquainted and talk things over in advance of the technical sessions. By evening a good party had assembled and Joe Brinton led the way to the ballroom for an informal reception and dance with punch and everything. A young bucko with dancing ideas but lacking in terpsichorean training did several rounds of toe dancing with the lady of my choice and I guess I trod the toes of some of the other wives.

On Tuesday some of the ladies got up early enough to play clock golf (half brother to the miniature variety) and for their putting skill in the face of tremendous obstacles, prizes were awarded to: Mrs. R. H. Carpenter, Mrs. F. G. Horton, Mrs. F. C. Houghten and Mrs. R. E. Daly.

After luncheon on Tuesday, though the skies were threatening, the crowd climbed into sightseeing buses and rode through Salem of witchcraft fame and then from Marblehead the steamer took them to Gloucester, home of the deep sea fishing fleet and the Gorton Fish Foundry, famous for dried cod and fish flakes. At Salem we saw the sights, House of Seven Gables, Gallows Hill and the Burial Place of Governor Bradford. The harbor at Marblehead was filled with small boats and the many quaint stores lining the narrow streets displayed ship models and early American antiques.—Not a good place to turn your wife loose if early American furniture is her hobby.

For those who liked the water in another form than supplied by a boat trip or swimming, the last few holes of the golf match left nothing to be desired. At Winchester Frank Tuttle turned the clam and divot diggers loose for a Research Session on the beautiful country club course, and when they finished exploring the traps, rough and woods and the last putt was poured into the cup at the 18th green, the Research Cup was in the custody of R. S. Franklin, Boston, who had blasted his way to victory over hill, dale and puddle for the low net score of the day.

In the evening after dinner the bridge-fiends started to work and at the end of a three-hour session Mrs. L. I. Carper and I. C. Jennings, carried off first honors and the other lucky ones were: Mrs. L. E. Seeley, Mrs. J. F. Tuttle, Mrs. W. T. Jones and Mrs. W. Webster, Jr., and Messrs. W. A. Swain, Warren Ewald, N. D. Adams and E. K. Campbell.

Wednesday was the big day and commenced with a sightseeing trip through Boston, visiting numerous historical spots, Bunker Hill Monument, Fanueil Hall, Boston Common and then to the Navy Yard for an inspection of the frigate *Constitution*, more affectionately known as Old Ironsides.

With perfect weather the golfers invaded Beverly and played a kickers' handicap tournament on the course of the United Shoe Machinery Co. The best guessers were: M. L. Steele, Robert Robertson, W. L. Fleisher, W. P. Brown, F. R. Ellis, T. J. Duffield, C. W. Lemmerman, J. J. Roper, C. J. Scanlan, L. I. Carper, W. C. Randall and W. R. Fraser.

After assembling at the big rock in front of the hotel for a convention photograph, the crowd marched into the Semi-Annual Banquet which was a red-hot repast of delightful dishes designed to tempt the most fastidious appetite. Each dish had been approved by the Council as you will see by the menu.

At the conclusion of dinner the Nottingham Quartet made a big hit with their vocal numbers and then Toastmaster W. T. Jones introduced J. F. Tuttle, Chairman of the Golf Committee who presented the Research Cup to R. S. Franklin, and read the list of prize winners for the Tuesday and Wednesday tournaments.

President W. H. Carrier was introduced and in a brief and appropriate speech presented Past President L. A. Harding, on behalf of the Council, with a handsome book of clippings, photographs and drawings to remind him of the important events that occurred during his administration.

The toastmaster then introduced as the speaker of the evening Captain Vermicilli Ravioli of the Italian Air Force and he delivered a brief and touching address in broken English while the assembled multitude sat breathless on the edge of their chairs. Upon being questioned as to what Mussolini thinks, it was revealed that the speaker (whose reply came in perfect English), was a well-known Bostonian who had never been in Italy and didn't speak Italian. (His dinner partners were greatly chagrined by this successful deception.)

At the conclusion of the banquet the boys and their girl friends adjourned to the ballroom where they stepped to the lively tunes of the New Ocean House Orchestra. There were no specialty numbers or prize dancers but an eminent Chicago jurist was unofficially awarded a handsome but in-

AMERICAN SOCIETY
OF
HEATING AND VENTILATING
ENGINEERS

SEMI-ANNUAL

BANQUET

Wednesday, June twenty-four
Nineteen hundred thirty-one

THE NEW OCEAN HOUSE
Swampscott, Massachusetts

Menu

Wenatchee Fruit Cocktail, *Eastwood*
Carpenter Saltines *Hoosier* Salted Nuts
Essence of Tomato, *Campbell*
Johnson Crackers, *McIntosh* *Larson* Hot Rolls
Lobster a la Newburg, *Cassellette*
Potatoes, *Boyden*
Stuffed Olives and Celery, *Gurney*
Filet Mignon, Chief *Carrier*
French Fried Sweet Potatoes, *Mensing*
Wax Beans, *Harding*
Romaine and Tomato, University *Rowley*
French Dressing, *Langenberg*
Vanilla Ice Cream with Crushed Strawberries, *Jones*
Assorted Cakes, *Rowe*
Demi Tasse, *Farnham*

visible trophy for having danced with the greatest number of ladies (at different times).

Early on Thursday morning the ladies assembled at the first tee of the New Ocean House Golf Course for a 9-hole tournament and the most proficient golfers were: Mrs. Thomas Chester, Mrs. F. C. Houghten, Miss Betty Driscoll and Mrs. J. W. Brinton.

The final event started under the guidance of T. (Transportation) F. McCoy at 2:00 p. m. Thursday when 70 members and ladies enjoyed a bus ride to Salem, Lexington and Concord and saw the historic monuments that marked the many places made famous in revolutionary days.

It was a delightful ending to a most successful meeting and I can't remember when I had such a good time. You should have been with us and I hope that after reading all of this you will mark your calendar now for the Annual Meeting in Cleveland next January.

Au revoir,
—V. E.

Record Breaking Heating and Ventilating Exposition Expected in Cleveland

THE Second International Heating and Ventilating Exposition, to be held January 25 to 29, 1932, in the Cleveland Auditorium, Cleveland, O., promises to be the most important and comprehensive event of its kind in the history of the industry according to information furnished the A. S. H. V. E. Advisory Committee which held its first meeting at Swampscott, Mass.

The personnel of the Committee is as follows:

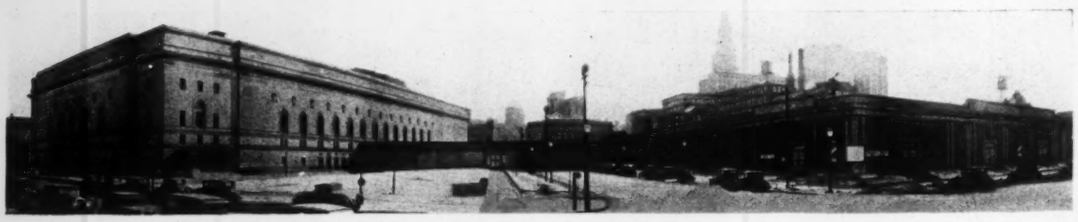
Walter Klie, *Chairman*
H. P. Gant, *Vice-Chairman*

D. S. Boyden	J. F. McIntire
A. C. Edgar	J. C. Miles
C. Gottwald	H. C. Murphy
C. V. Haynes	M. F. Rather
C. H. B. Hotchkiss	E. A. Scott
W. C. Kammerer	W. E. Stark
F. P. Keeney	F. R. Still
F. A. Kitchen	T. A. Weager

A. W. Williams

So far over 60 per cent of the available space in the Cleveland Auditorium Annex has been contracted for and this volume of advance sale exceeds the amount reserved during the corresponding period for the first show held in 1930.

The plan of booths is unique and will impress the visitor entering the main gate. Directly in the center will be the heating and ventilating section flanked by the gas burning section, the oil burning section, the warm air heating section and the refrigerating and air conditioning division. A large exhibit of domestic stokers burning hard and soft coal will be a feature, and all of these units together with oil and gas burning equipment will be in actual operation. The demonstration of equipment should have a special appeal to members and guests of the American Society of Heating and Ventilating Engineers and the *American Society of Refrigerating Engineers,* will hold their Annual Meetings in Cleveland at the same time as the Exposition. Thousands of people will come from communities accessible to Cleveland and the record attendance of 57,000, who visited the first exposition in Philadelphia 1930, should be surpassed. Cleveland will be the capital of the heating and ventilating world during the fourth week of January, 1932.



Cleveland Auditorium

The following exhibitors have contracted for space:

Aerofin Corp.
Aerologist, The
Air Reduction Sales Co.
American Air Filter Co.
American Gas Association
American Gas Products Co.
American Oil Burner Assn.
American Radiator Corp.
American Society of Heating and Ventilating Engineers
American Society of Refrigerating Engineers
American Temperature Indicating Co.
Ames Iron Works
Ames Pump Co., Inc.
Armstrong Machine Works
Armstrong Cork Co.
A. G. A. Testing Laboratory
Automatic Burner Corp.

B-Line Boiler Co.
Barber-Colman Co.
Barber Gas Burner Co.
Barnes & Jones
Bishop & Babcock Sales Co.
Brownell Co., The
Bryant Heater & Mfg. Co.
Buckeye Blower Co.

Carrier Corp.
Carrier Engineering Corp.
Carrier-Lyle Corp.
Century Electric Co.
Century Engineering Corp.
Chicago Pump Co.
Cleveland Gas Burner & Appliance Co.
Cleveland Steel Products Corp.
Cook Electric Co.
Cooling and Air Conditioning Corp.
Coppus Engineering Corp.
Crane Co.
Crystal Oil Burner Corp.

Davison Gas Burner & Welding Co., N. C.
Domestic Engineering
Dunham Co., C. A.
Struthers Dunn, Inc.

East Ohio Gas Co., The
Economy Pumping Machinery Co.
Emerson Electric Mfg. Co.
Engineering Publications, Inc.
Estate Stove Co., The

Fox Furnace Co.
Fuel Oil Journal

Gas-Aire, Inc.
General Electric Co.
General Gas & Light Co.
Grinnell Co., Inc.

Hart Oil Burner Corp.
Heating Journals, Inc.
Heating, Piping & Air Conditioning
Heating & Ventilating
Henry Furnace & Foundry Co., The
Hill Co., E. Vernon
Hoffman Specialty Co., Inc.

Illinois Engineering Co.
Iron Fireman Mfg. Co.
Independent Air Filter Co.

Johnson & Co., S. T.
Johnson Service Co.
Johns-Manville Corp.

Kewanee Boiler Corp.
Kieley & Mueller, Inc.

Leeds & Northrup Co.
Linde Air Products Co.

McCord Radiator & Mfg. Co.
McDonnell & Miller
Marsh & Co., Jas. P.
May Oil Burner Corp.
Mercoid Corp.
Mineral Felt Insulating Co.
Minneapolis-Honeywell Regulator Co.
Motor Stoker Corp.
Mouat Vapor Heating Co.
Mueller Furnace Co., L. J.
Multicell Radiator Corp.

Nash Engineering Co.
National Radiator Corp.
Nelson Corp., Herman

Oil Heat
Oxweld Acetylene Corp.
Oil Heating Institute

Parker Appliance Co.
Peerless Unit Ventilation Co.
Penn Electric Switch Co.
Pennsylvania Furnace & Iron Co.
Pierce Butler & Pierce Mfg. Co.
Piqua Electric Mfg. Co.
Powers Regulator Co.
Propellair, Inc.

Raymond Co., F. I.
Reif-Rexoil Co., Inc.
Richardson & Boynton Co.
Robertson Co., H. H.
Rochester Circulator Sales Co.
Russell & Co., W. A.

Sarco Co., Inc.
Silent Automatic Corp.
Silent Glow Oil Burner Corp.
Spencer Heater Co.
Stay-Rite Co., Inc.
Sun Radiator Cover Co., Inc.
Superior Steel Corp.
Surface Combustion Co.
Swartwout Co., The

Taco Heaters, Inc.
Time-O-Stat Controls Co.
Trerice Co., H. O.

Union Carbide Co.
United States Radiator Corp.

Warren Webster & Co.
Warm Air Furnace Fan Co.
Weil-McLain Co.
Whittington, Inc., W. P.

York Heating & Ventilating Corp.
York Ice Machinery Corp.
Young Radiator Co.
Youngstown Sheet & Tube Co.

Electrical Engineers Elect Officers

Charles E. Skinner, assistant director of engineering, Westinghouse Electric and Manufacturing Company, East Pittsburgh, Pa., was elected president of the *American Institute of Electrical Engineers* for the year beginning August 1, 1931, at the meeting of the Institute held at Asheville, N. C., June 22. The other officers elected were: Vice-presidents W. B. Kouwenhoven, Baltimore, Md.; W. E. Freeman, Lexington, Ky.; Paul H. Patton, Omaha, Nebraska; A. W. Copley, San Francisco, Calif.; L. B. Chubbuck, Hamilton, Ontario.—*Directors* L. W. Chubb, East Pittsburgh, Pa.; B. D. Hull, Dallas, Texas; H. R. Woodrow, Brooklyn, N. Y.—*National Treasurer* W. I. Slichter, New York, N. Y. (reelected).

These officers, together with the following holdover officers, will constitute the Board of Directors for the next administrative year, beginning August 1: W. S. Lee (retiring president), Charlotte, N. C.; Harold B. Smith, Princeton, Mass.; H. V. Carpenter, Pullman, Washington; G. C. Shadd, Lawrence, Kansas; I. E. Moultrop, Boston, Mass.; H. P. Charlesworth, New York, N. Y.; T. N. Lacy, Detroit, Mich.; J. Allen Johnson, Buffalo, N. Y.; A. M. MacCutcheon, Cleveland, Ohio; A. E. Bettis, Kansas City, Mo.; J. E. Kearns, Chicago, Ill.; F. W. Peek, Jr., Pittsfield, Mass.; C. E. Stephens, New York, N. Y.; A. B. Cooper, Toronto, Ontario; A. E. Knowlton and R. H. Tapscott, New York, N. Y.

House Heating Plants and Smoke Prevention

At the 25th annual convention of the *Smoke Prevention Association of the United States*, held June 30 to July 3, in Grand Rapids, Mich., V. W. Cherven of Holland Furnace Co., Holland, Mich., delivered a paper on Smoke Abatement with Warm Air Furnaces. In this paper he discussed the functions of the chimney flue and pointed out that 80 per cent of heating complaints are traceable to inadequate and leaky chimneys.

The necessity of keeping the heating plant clean and having periodical inspection was emphasized and the fundamental principles of proper firing of central heating plants were briefly outlined.

The necessity for educational work through cooperation of engineers, contractors and manufacturers was urged as a means of preventing air pollution by residence heating plants.

Carnegie Tech Appointments

Carnegie Institute of Technology announces the appointment of Clark M. Humphreys as assistant professor of plumbing, heating and ventilating, effective with the beginning of the next school year. He will be in charge of the curriculum of this department under the Theodore Ahrens Professorship.

Professor Humphreys is a graduate of the Ohio State University where he studied mechanical engineering. For the past six years since graduation he has been in charge of the heating and ventilating and mechanical equipment work for the State Architect's Office of Ohio at Columbus.

In his work at Carnegie Tech Professor Humphreys will be assisted by Theodore F. Rockwell, who has been appointed instructor in plumbing and heating. Mr. Rockwell has just completed his work under a research fellowship sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The study was made at the University of Minnesota where he was awarded the master of science degree. His practical experience has been in the field of plumbing, piping and general engineering construction.

Committee of Ten Officers Reelected

As a result of a mail ballot the Committee of Ten for the Coal and Heating Industries have re-elected officers for the coming year as follows: *Chairman*, H. A. Glover, New York, representing the *National Coal Association; Vice-Chairman*, E. B. Langenberg, St. Louis, representing the *National Warm Air Heating Association; Treasurer*, H. H. Kurtz, Chicago, representing the *Stoker Manufacturers Association* and the *Midwest Stoker Association; Secretary*, L. W. Smith, Jr., Goshen, Ind., sales promotion specialist for coal heating appliances and accessories.

A luncheon meeting of the Committee was held in Chicago at the Union League Club, July 22, and matters for special subcommittee action were discussed.

Society Cooperating With the President on Home Building and Home Ownership

AN ORGANIZATION meeting of the Committee on Fundamental Equipment for the President's Conference on Home Building and Home Ownership was held in Washington, D. C., July 17, to consider problems of mechanical requirements for the home and Prof. C. P. Bliss of New York University, Chairman of the Committee, outlined the scope of the Committee's work which is to prepare recommendations on heating and ventilation, humidification, refrigeration, plumbing, lighting systems, so that the standards of living in the United States may be further advanced and broadened.

A special Committee to consider heating, ventilating and air conditioning problems was appointed with the following A. S. H. V. E. members: Prof. A. C. Willard, *Chairman,* Urbana, Ill.; W. H. Driscoll, New York; Philip Drinker, Boston, Mass.; and Perry West, Newark, N. J. and they will be assisted by Prof. M. L. Nichols, Auburn, Ala. and Dr. Jesse Churchill, New York.

SUB-DIVISIONS OF COMMITTEE ON FUNDAMENTAL EQUIPMENT

1. *Heating and Ventilation and Air Conditioning* (all systems) including humidification, washing, drying and circulation of air.
   - (*a*) Fire places, stoves and chimneys.
   - (*b*) Fuels, boilers and heat producing equipment (including oil, gas, coal, electricity, etc.).
   - (*c*) Automatic Controls.
   - (*d*) Ventilating the home (including methods, equipment and health considerations).
   - (*e*) House cooling equipment.
2. *Plumbing and Sanitation.*
   - (*a*) Systems, urban and rural (including drinking water, rain water and its collection and disposal).
   - (*b*) Installation and fixtures.
   - (*c*) Care and repair of installations.
3. *Lighting* (including gas, electricity, acetelyne).
   - (*a*) Wiring and outlets location, systems and safety.
   - (*b*) Fixtures—permanent and portable (including indirect and color lighting).
   - (*c*) Daylight distribution (confer with Committees 10 and 11).
4. *Refrigeration and Cooling* (including ice, electricity, gas and other methods).
   - (*a*) Refrigerators—types, cost and performance (including refrigerants and hazards).
5. *Built-in Equipment.* Laundries, garages, furniture, fire escapes, radios, house telephones, dumb-waiters and elevators (consult overlapping committees, such as 10 and 11).
6. *Public Inspection and Certification.* Methods applicable to all types of home equipment.
7. *Publicity* (Methods of presentation).
   - (*a*) Technical knowledge to special groups concerned.
   - (*b*) Advice to home builders, buyers and owners.

FUNDAMENTAL EQUIPMENT COMMITTEE

List of Members

George B. Bright (Refrigerating Engineers and Architects) George B. Bright Co., 2615 12th St., Detroit, Mich.

George K. Burgess, (Physicist) Director, Bureau of Standards, U. S. Department of Commerce, Washington, D. C.

E. F. Carter, Director, Division of Standards and Inspections, Raleigh, N. C.

Dr. Jesse Churchill, The Chemists' Club, 52 East 41st St., New York City.

W. H. Driscoll, (Building Construction) Vice-Pres., Thompson-Starrett Co., 250 Park Ave., New York City.

A. E. Hansen, (Consulting Engineer) 1350 Broadway, New York City

Mrs. Paul E. Howe, American Home Economics Assn., Mills Building, Washington, D. C.

Robert T. Jones, Architects' Small House Service Bureau, 1200 Second Ave. S., Minneapolis, Minn.

Matthew Luckiesh, Director, Lighting Research Laboratory, General Electric Co., Nela Park Engineering Dept., Cleveland, Ohio.

J. L. Murphy, President, J. L. Murphy, Inc., (General Piping Contractors), 350 East 44th St., New York City.

Prof. Mark L. Nichols, Head Department of Agricultural Engineering, Alabama Polytechnic Institute, Auburn, Ala.

Roy A. Palmer, (Tentative) (Illuminating Engineer), Southern Public Utilities Co., Charlotte, N. C.

Dr. Louise Stanley, Chief, Bureau of Home Economics, U. S. Dept. of Agriculture, Washington, D. C.

Perry West (Consulting Engineer), 13 Central Ave., Newark, N. J.

Prof. A. C. Willard, University of Illinois, 1208 West California St., Urbana, Ill.

Each of the sub-divisions coming on to the Committee on Fundamental Equipment, will be in charge of a special group, who will make a study of existing conditions and prepare a report for consideration at the President's Conference on Home Building and Home Ownership to be held in Washington, D. C., during November, 1931.

COMMITTEES FOR THE PRESIDENT'S CONFERENCE ON HOME BUILDING AND HOME OWNERSHIP

1. *Types of Dwellings*
   (a) Single family houses; (b) Two-family houses; (c) Multiple dwellings.
2. *Fundamental Equipment* (Heating, Lighting, Plumbing, Ventilation and Air Conditioning)
3. *Kitchen and other Work Centers*
4. *Utilities for Houses*
5. *Subdivision Layout*
6. *Business and Housing* (Past efforts and future programs)
7. *Industrial Decentralization and Housing*
8. *Blighted Areas and Slums*
9. *Reconditioning, Remodeling, and Modernizing*
10. *Construction* (Materials, methods, etc.)
11. *Design*
12. *City Planning and Zoning*
13. *Finance*
14. *Taxation*
15. *Home Ownership and Leasing* (or Renting)
16. *Home Furnishing and Decoration*
17. *Landscape Planning and Planting*
18. *Household Management*
19. *Housing and the Community* (Relation to health, education, recreation, citizenship, industrial efficiency and delinquency)
20. *Farm and Village Housing* (The isolated farm house and the village)
21. *Negro Housing* (Racial, National or Alien groups)
22. *Home Information Centers*
23. *Homemarking*
24. *Large Scale Operations* (Planning, building and management of property)
25. *Relationship of Income and the Home*

CORRELATION GROUP

A. Standards and Objectives
B. Research
C. Legislation and Administration
D. Education and Service (Home Information Centers)
E. Organization Program, Local and National
F. Technological Developments

Kewanee Moves Detroit Office

The Kewanee Boiler Corp., Kewanee, Ill., announces that on July 20, the Detroit branch office was moved to new quarters at 1344 Broadway, where B. H. Schulze, manager and his staff are ready to welcome their friends.

New Engineering Offices Opened

An announcement from Albert B. Franklin, Inc., heating and ventilating engineers and contractors since 1878, tells of the establishment of engineering offices at 38 Chauncy St., Boston, Mass., the contracting department having been discontinued.

Boston to Entertain Oil Burner Men in 1932

Announcement has been made by the Board of Directors of the *American Oil Burner Association* that the 9th Annual Convention and Oil Burner Show is to be held in Boston, Mass., during the month of April, 1932.

Oil Burner Association Appoints Committees

Announcement has been made of the appointment of several new standing committees by the *American Oil Burner Association,* 342 Madison Ave., New York, N. Y., as follows:

Executive Committee: Walter F. Tant, *Chairman,* Haldeman Finnie, E. M. Fleischmann, W. J. Smith, Earl Marr.

Committee on Cooperation with Heating Industry: J. P. McCarthy, *Chairman,* Edward J. Wohlfarth, Homer Linn, G. A. Markuson, J. T. McTarnahan and Cliff W. Presdee.

Technical Research Committee: John H. McIlvaine, *Chairman,* H. J. G. Rudolf, T. H. Smoot, R. W. Beckett, A. J. Frame, W. C. Schoenfeldt.

The Technical Research Committee is cooperating with the A.S.H.V.E. in investigations of oil burner and boiler performance at Yale University, New Haven.

Kansas City

June 15, 1931. The annual spring party given at Quivera Lakes Club House near Kansas City, was a fitting climax to a successful year of the Kansas City Chapter. The 85 present enjoyed the swimming and boating which preceded the dinner. The beautiful table decorations were arranged by Mrs. E. K. Campbell, and the flowers were furnished from her own gardens. Wendover's nine piece orchestra furnished music during the dinner and for dancing later in the evening.

Routine business was disposed of quickly, and the new officers and board of governors duly elected as follows:

President—J. M. Arthur.

Vice-President—W. A. Russell.

Secretary—D. D. Zink.

Treasurer—Emil Haas.

Board of Governors—E. K. Campbell, T. L. Dawson and J. G. Lewis.

Professor Cady of Kansas University gave an interesting talk and demonstration on Liquid Air.

Death of John R. Weaver

While driving east J. R. Weaver died in Arizona on June 5. Mr. Weaver was an active and enthusiastic member of the Southern California Chapter at Los Angeles and for the past six years had been with O. W. Ott, consulting engineer, Los Angeles.

In his entire business career Mr. Weaver had specialized in heating, ventilating and power plant work. His first position after leaving school in 1912 was with Charles Brossmann, consulting engineer at Indianapolis. From 1916 to 1922 he was located in Pittsburgh with the Crucible Steel Co. of America and the Carnegie Division of the U. S. Steel Corp.

Mr. Weaver went to California in 1922 and was associated with Allison and Allison, architects, and then with D. S. Reynolds in engineering work. Mr. Weaver was born in Indianapolis, Ind., January 6, 1880, and received his education in the public schools of that city.

It is with regret that the Officers and Council learn of his untimely death and their sincere sympathy is extended to his family and associates for the loss that they have sustained.

Death of Robert W. Pryor, Jr.

The heating and ventilating industry lost one of its well-known and universally liked members in the death of Robert Westall Pryor, Jr., who passed away July 10 at his home in Newark, N. J., after an illness of several months. Mr. Pryor was in his 50th year having been born July 31, 1881, at Newark, N. J., where he spent practically his entire lifetime.

After preparing for a career in mechanical engineering at Stevens School, Hoboken, N. J., he graduated from Stevens Institute of Technology in 1902 with the degree M.E. Immediately upon graduation he entered the employ of Buffalo Forge Co., Buffalo, N. Y., where he gained drafting and shop experience before doing engineering and sales work for the company. For

Robert Westall Pryor, Jr.

many years he has represented the company in the New York territory and from 1909 as a partner in the firm of Koithan and Pryor, who were responsible for numerous applications of heating and ventilation and drying systems in the metropolitan district.

Mr. Pryor's life interest was in engineering and he had a host of friends in the heating and ventilating profession. He was of genial disposition and his particular outdoor hobbies were camping, hunting, fishing and golf and many stories are told by his intimates of his prowess as a camp cook.

Mr. Pryor joined the Society in 1913 and served a two year term on the Council during 1919 and 1920.

The officers and members of the Council feel a great loss in the death of Mr. Pryor and extend their sincere sympathies to his widow, Lillie K. Pryor, and his daughter, Mary D. Pryor. His passing will also be mourned by his many friends in the organizations to which he belonged, *A. S. M. E.,* Engineers Club of New York, Dotoc Athletic Club and the Essex Falls Country Club.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 7 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by August 14, 1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
CASEY, HUNTLEY FANT, C. M. Guest & Sons, Anderson, S. C.	F. E. Baird F. W. Claire	E. W. Klein C. L. Templin
CRANSTON, WILLIAM EMER, JR., Vice-President, Hoffman Specialty Co. of California, Pasadena, Calif.	G. D. Hoffman W. K. Simpson	N. P. Fenner R. E. Jones
HITCHCOCK, PAUL CAMERON, Pillsbury Engrg. Co., Minneapolis, Minn.	H. E. Gerrish Albert Buenger	E. B. Gordon, Jr. W. F. Uhl
HJERPE, CLARENCE A., JR., 93 Winthrop St., New Britain, Conn.	Prof. S. E. Dibble E. H. Riesmeyer, Jr.	E. A. Stewart G. R. Dahlman—*Non-Member*
MARSH, THOMAS A., Modern Coal Burner Co., Chicago, Ill.	E. P. Rich—*Non-Member* W. L. Fergus—*Non-Member*	H. R. Linn Edgar Nethercutt—*W.S.E.*
MESSENGER, JOHN R., Manager, Ventilating Dept., Universal Sheet Metal Co., Los Angeles, Calif.	H. B. Keeling C. E. Johnson	C. S. Anderson O. W. Ott
VINCENT, PAUL JOSEPH, Chatard & Norris, Baltimore, Md.	O. W. Armspach W. F. Christman	G. D. Fife C. N. Witmer

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

COOPER, JOHN RAYMOND, Eastern Branch Manager, Thermal Units Mfg. Co., New York, N. Y.

GASSLER, JOHN H., Manager, Engrg. Dept., Crane Co., Jacksonville, Fla. (*Advancement*)

GROSSMAN, HARRY A., Owner, H. A. Grossman Co., St. Louis, Mo.

HVOSLEF, FREDRIK WALDEMAR, Kohler Co., Kohler, Wis. (*Reinstatement*)

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SHIMANSKI, VICTOR EDWARD, Sales Engr., Trane Co., La Crosse, Wis.

WATERS, GEORGE G., District Manager, American Blower Corp., Pittsburgh, Pa. (*Advancement*)

ASSOCIATES

GIRRBACH, GEORGE FREDERIC, McQuay Radiator Corp., Minneapolis, Minn.

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Heating·Piping and Air Conditioning EDITORIALS

A Debt of Gratitude

In a paper presented before a recent meeting of the *A. S. T. M.* and the *A. S. M E.*, it was suggested by the author, F. W. Martin, that the range of operating temperatures above 750 F be divided into three bands; 750 to 850 F, 850 to 1,000 F and temperatures over 1,000 F. Mr. Martin's paper is published in full elsewhere in this issue.

According to Mr. Martin, this is advisable from the standpoint of standardization, simplicity of design, and reduction of manufacturing costs. With this as a basis, he further states, there will be one set of standard dimensions with one set of dies and patterns for all pressures and four grades of material depending on the temperature.

Recently, a consulting engineer stated "It is a compliment to the engineering profession that after the conditions arose incidental to the application of higher pressures and temperatures, the development of new alloys has made such application possible. Engineers have, for a long time, been aware of the economics of such practice, but only in recent years have the advantages been fully appreciated. It is interesting to note that there are few new features in design, but rather a refinement and rigidity in the requirements usually assumed."

The piping engineer owes a debt of gratitude to the metallurgist, who has made so much possible through the development of new and improved alloys. There is a constantly growing need for reliable data on the performance of materials for high temperatures and high pressure piping, which is being supplied by research laboratories and operating tests from many forward-looking organizations

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Fans as Stage Properties

We went to a neighborhood movie last night, to see one of the almost-extinct aviation pictures. During the usual dog-fight, where the inexperienced hero nearly succeeds in bringing down the "Red Colonel" someone speeded up the ventilating fans. A gale of air streamed past the audience from the grilles in the front wall of the building. The noise of the fans added to the whir of the propellers only served to make the play more real. But later, when the action switched to conversation, the fans kept right on whirring.

We object to theater managers using the air conditioning and ventilating equipment as stage properties. Surely, its function as a comfort provider is important enough to justify its cost without making it "double in brass."

And another thing: We heard the other day of a manager who ordered his engineer to "give 'em all the cold air you can." The air conditioning system was well-designed, and could be operated to balance the outdoor conditions. But with oceans of cold air filling the interior with no regard to the people who had just come in from the sweltering streets, there must have been many who doubted the benefits of air conditioning.

Proper operation of air conditioning systems is extremely important. It should be left to the engineer in charge.

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Serving Mankind

It has been rumored lately that one automobile company has designs for a new model embodying many departures from the conventional. Among other improvements, it is said to provide for air conditioning and cooling the air.

Air conditioning is taking to wheels, evidently, much as the proverbial duck takes to water. Last month you will recall the pictorial feature published in this magazine describing the Baltimore & Ohio's air conditioned "Columbian"; advice recently received from the same railroad announces that two more trains have been completely conditioned. Other roads are running air conditioned diners, and are finding them business-pulling attractions.

Buildings are built without any windows—but they are air conditioned. Other structures are constructed having walls practically entirely of glass—they are air conditioned also. A large office building recently put up in the middle west was not equipped with a conditioning system—it has been found necessary to install air conditioning for several areas in this structure.

Perhaps we're prejudiced; perhaps we know too much about what air conditioning can do for industrial plants, for office buildings, for hotels, theaters, and a myriad other types of enclosures. But even allowing for that it would seem that air conditioning

is the industry of today and of the future—that there is hardly any type of structure which is complete without air conditioning—that the air conditioning engineer is serving mankind as few professions have done before.

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Warm Air Heating

If ever there was a time when the warm air heating engineer should put his best foot forward and make a bid for his rightful place in the heating of larger structures, it is right now. In earlier days, all the heating man thought of was the production of heat. Later he was compelled to consider heat and humidity. Then cleanliness, motion and ionization of air arose as legitimate accompaniments of heating, the whole developing—with cooling—into what is now termed air conditioning. The entire country is becoming "air conditioning minded" as advertising men term it. We want the best there is in air, as we want the best there is in everything else. In the meantime, air conditioning for large buildings has jumped ahead of its "foster parent," warm air heating. The relationship readily can be traced, for a modern air conditioning system with its elaborate equipment of ducts, fans, air filters, etc., is perhaps more like the time honored warm air system than it is like any other type of heating. And the warm air man who fails to see and appreciate this analogy and to perceive the possibilities of development in the near future in the warm air field, is, we believe, a little shortsighted.

When warm air furnaces are used in buildings where temporary or intermittent heat is required, their ability to heat quickly is valued, as in garages when there is a sudden fall in temperature, in churches where the occupancy is intermittent, in schools where the buildings are not needed during the greater part of the 24-hour day, etc.

The modern furnace with its fan, automatic humidifier and air washer is, however, following closely in the footsteps of the complete air conditioning system. What the future development will be one can scarcely predict.

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Refrigeration and Air Conditioning

Theater cooling is commonly spoken of as air conditioning. It should not be forgotten, however, that the refrigerating engineer is an important factor. Air conditioning for comfort and health means control of humidity and for three or four months of the year dehumidifying is necessary to remove the excess of moisture from the air. Dehumidifying often is accompanied by refrigeration. For this reason in any study of the future of refrigeration, consideration must be given to the demands of air conditioning. In its use here the word "refrigeration" is outside that of the meaning of the word as often used because in theaters it is installed solely for cooling and has no relation to ice making purposes. Air conditioning where refrigeration is used is not primarily cooling but the accurate control of air motion, temperature, humidity and cleanliness.

More than 300 theaters are now operating air conditioning systems. It is estimated that the average refrigeration tonnage is 60 tons per thousand seats, and that there are 12,000 to 15,000 additional theaters for which the refrigerating equipment alone would cost $100,000,000.

Refrigeration is a great factor in the box office receipts in all air conditioned theaters in the United States. And when patrons go to the air conditioned theater in preference to the one that is not air conditioned, it is also an important factor in the box office receipts of the theater that is not air conditioned.

This, of course, is only one outlet for refrigeration tonnage. There are many other obvious outlets in department stores, hotels, etc. The president of a refrigerating society told its members, "Gentlemen, do not lose sight of the opportunities that are rapidly developing in the air conditioning field."

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No Absolute

Dr. C. E. Kenneth Mees has been the cause of several columns of "dissenting opinions" in the newspapers, replying to his recent criticism of engineering, in which he said that the engineer had not benefitted mankind, save in checking disease. It is interesting to note that, as in many questions, the arguments for and against depend, of necessity, on the definition or ultimate meaning of a word.

We venture to say that there are probably as many meanings ascribed to "benefit" as there are persons using the word. For instance, to Dr. Mees the normal life of man is an agricultural one, according to his statements. But we believe that there are many farmers who look upon the life of a city dweller as man's normal mode of existence. Dr. Mees condemns engineering for leading the way—or, at least, making the way possible—away from his idea of man's normal life.

Dr. Mees' paper was of considerable interest to us, and we are sure every engineer will find it so; it is published in this issue. A compilation of data and "case histories" supporting and opposing his viewpoint would also be of more than passing interest and value. But words and meanings being what they are, such a compilation would be extremely difficult; it is regrettable that "benefit" has no absolute meaning, unaffected by the personality of the user.

Practical Piping Problems

Care of Valve Stems—Opening "Stuck" Valves

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

The stem is one of the important parts of the valve, as judged by the operating and maintenance men. Broken valve stems mean delays and trouble. Broken stems are, in some cases, caused by careless handling of the valves. In other cases it is sometimes necessary to take a chance in operating a valve and even with good judgment and careful operation a broken stem may be the result.

Valves of the rising stem pattern with outside screw and yoke have an excellent feature in that the position of the stem is an indicator as to whether the valve is closed or to what extent it is opened. The outside screw stem may be a detriment when the valve is installed where the stem is subject to corrosion. A valve with a stem of this type often is not desirable when located in an underground pit, where moisture and dirt will come in contact with the threaded stem, especially when the valve is not operated frequently.

Care of Valves Necessary

Valves should be kept well packed, operated at periodical times and stems oiled and lubricated, when necessary. Operating conditions and various other duties do not permit this work being done, in the majority of plants, both large and small. The valves, in many cases, are left to take care of themselves, especially when installed in pits, tunnels and places not readily accessible.

It is not an unusual occurrence for an employe, when inspecting valves of the stationary stem, inside screw type, to report a valve open, when it is closed or closed when it is open or partly open. Cases have been experienced where a valve, when in the wide open position, could not be readily operated, was assumed to be closed and the stem was broken by trying to force the handwheel in the direction for opening the valve.

Opening Valves Which Are "Stuck"

Supervisors of maintenance work occasionally receive a report from an employe that a certain valve cannot be closed, or opened, the handwheel and stem being apparently immovable. Valves in this condition should

be treated carefully. When it is not certain whether the valve is open or closed and there is no visible means of determining the position of the valve, try to move the handwheel first in one direction and then in the other; experience is the only practical way to tell how much power can be applied. If the valve stem does not move when ordinary power is applied and it is necessary to use a wrench, it is better to apply the power from opposite sides of the handwheel at the same time, in order to prevent a side strain on the stem.

If this method fails to start the valve stem moving and the valve is accessible, loosen the packing box gland as much as pressure conditions will permit and pour oil —penetrating oil is suitable—around the stem, apply a steady, even pressure to the handwheel, first in one direction and then in the other. While this power is being applied to the handwheel, strike the body of the valve with a heavy block of wood. It may take time and patience, but it is best to proceed carefully. This method has been successful in many cases.

Opening a Water Valve

Occasionally underground water gate valves are hard to operate. Recently a case was brought to the writer's attention of a valve on an underground water line that had not been closed for a number of years. The valve was reported as out of order and apparently could not be closed. As a test was being made on the water lines, it was necessary to have this valve closed. After carefully pulling on the valve key wrench, first in one direction and then in the other, the valve stem was started. After an hour's work of this kind the valve stem was moved four turns. The work was continued carefully and patiently until the valve was closed, moving the valve stem thirty-nine turns. The lower part of the interior of the valve body contained dirt and scale. By alternately closing and opening the valve a few turns, with a flow of water through the pipe line, the valve was flushed out and closed tight against the water pressure.

TESTING A GATE VALVE STEM AT THE UNDERWRITERS' LABORATORY, CHICAGO

Care Prior to Installation

Careful handling of valves prior to being installed is essential. Many times large valves are thrown off of trucks, causing the stems to be bent and handwheels broken. Valves should be kept tightly closed, in handling, prior to being installed, in order that sand, dirt and other foreign matter does not remain in contact with the seating faces of the discs and gates. Special attention should be given to valves of the rising stem type, in this respect, as the stem extends outward as the valve is being opened. Careless handling may result in a bent stem, or the handwheel of the valve may be broken. A handwheel on this type of a valve is not replaced easily.

Capable maintenance pipe fitters, when selecting valves, will probably look first to note the kind of seat and disc or the seating faces of the gate valves, but a strong, well made valve stem, of good material, securely attached to the disc, or the gate of the valve, will not pass unnoticed.

Joints in New Line Steam Cut

A subscriber asks why the joints in a new 125-lb. line should become steam cut. This may be due to several causes.

Referring to page 273, HEATING, PIPING AND AIR CONDITIONING for March, illustration *B*, a pipe thread is shown that was removed on account of a groove cut in the thread caused by the steam and water of condensation from the steam leaking through this joint.

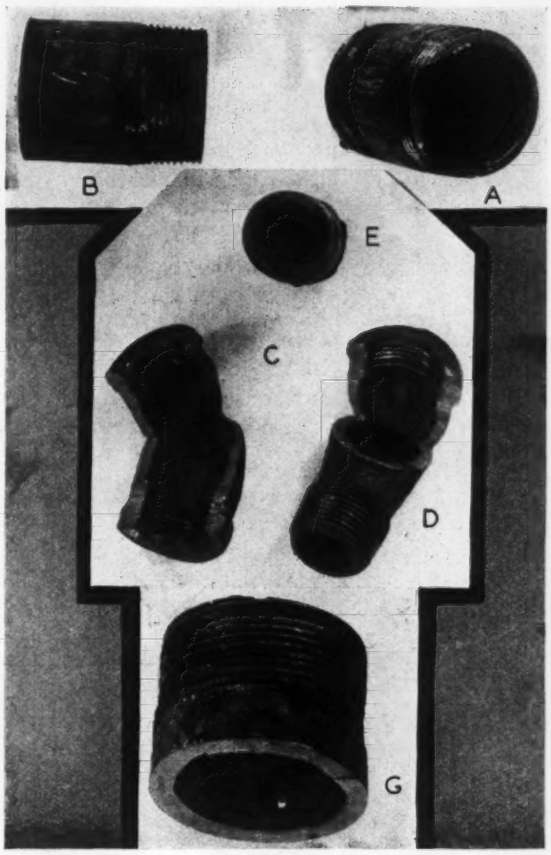
Causes of Leaks

Experience with leaks of this kind has shown that they may be caused by:

1. Insufficient pipe joint compound or lubrication applied to the threads, both male and female, when the pipe joint is made up; compound not spread evenly and carefully over all of the contact threads. Threaded pipe joints made up dry or partly dry are almost sure to leak and cause trouble when pressure is put on the pipe line.

2. Threads on the pipe not cut properly; broken threads, and threads not having the proper taper, consequently not engaging properly with the threads in the fittings. There have been cases where oil used for the pipe threading tools and machines was not suitable for the purpose, causing the threads to tear out in places. Improperly adjusted dies for pipe threading tools will not cut the regular pipe thread with the standard taper. This may occur when dies for threading machines are ground in resharpening. If some of the teeth in the chasers are ground off, the thread will be a straight thread for part of the length and will not mesh properly with the thread in the pipe fitting.

3. Threaded joints not made up tight. Threaded pipe joints should be screwed up tight, though care must be taken not to force the threads and strain the fittings



A Shows a Nipple the Interior of Which Is Worn.
B Shows a Pipe Thread Which Was Cut on the Outside by Leaking Steam.
C Shows an Elbow Removed from a Steam Line.
D Shows a Street Elbow Removed from a Steam Line.
E Shows a ¾-in. Pipe Nipple Removed from a Pump Discharge Line with Water at Atmospheric Temperature.
G Shows a Piece of 2½-in. Pipe Removed from a Boiler Feedwater Line.

Many leaks develop in pipe lines after being in service a short time on account of failure to make tight pipe joints.

4. Insufficient hangers and supports in the pipe line to take the weight and strain.

5. Steam lines not provided with traps, drains or other means for keeping the pipe line clear of condensation. A steam line that is allowed to fill up or contain a large amount of condensation at times, causing intermittent temperature changes, is liable to develop leaks in threaded and flanged joints. Shocks and strains, due to water hammer, may loosen the joints and cause leaks.

6. Steam lines not provided with expansion joints or other means for compensating for expansion and contraction. In some cases the absence of expansion joints allows a strain to be put on the pipe line joints, which causes leakage.

Leaks caused from threads being cut can be repaired, in many cases, by taking the threaded pipe joint apart, cleaning the threads and seeing that they are free from rust and from all old pipe joint compound. Then a coating of special compound (mixed according to the manufacturer's instructions) should be applied to the male and female threads of the pipe joint and the joint screwed up tight before the special compound sets and hardens.

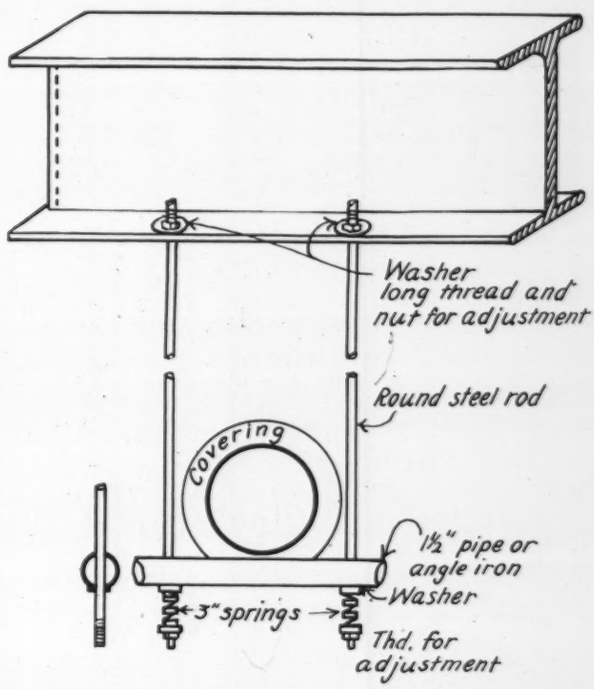
Pipe Supports

It was a privilege to read the article "Supports and Anchors for Piping" in the May issue. The subject appears timely and covers pertinent points of pipe engineering, that are often lost sight of in the general layout of a power plant.

Might the writer suggest in this connection that it has been found advantageous in some designs to attach the steam separator directly and solidly to the foundation of the engine and continue from there to the port opening, thus transferring the entire piping load or weight from the throttle valve and engine frame.

The correct alignment of all fittings to obviate the necessity of drawing or throwing strains on one side of the flange is an essential factor to good fabrication practice.

Too much care cannot be given the field work erection of piping to have easy practical, mechanical fitting of joints. It is often considered good practice to provide individual steam separators and accessible strainers to each power unit. This point can also be reckoned from an insurance point of view, for clean, dry steam is quite imperative for the smooth, economical running of an



This Support Holds the Pipe Securely and Assures a Cushion

engine and to insure efficient use and prolonged life to steam nozzles and impellers.

Anchors and pipe supports are vital parts of a smooth working job. The diagram shows a detail of a support that frequently is used where the steam main comes from over head and the final connections are not as flexible as may be desired. This type of support offers not only a secure means of holding the pipe, but also assures a cushion.—*T. M. Dugan.*